COMPUTER AIDED DESIGN OF MACHINE ELEMENTS

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in

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By

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ABSTRACT

COMPUTER AIDED DESIGN
OF MACHINE ELEMENTS

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In this thesis, a computer aided design (CAD) package for the machine elements is prepared. Shafts, bolts, anti-friction bearings, journal bearings, helical springs, brakes and V-belts are the elements considered in this study.

The developed design package has a modular structure. For each of the machine elements specified above, an interactive computer aided design/selection program is written. A database which consists of material properties, design parameters like stress concentration factors, charts, standard values from manufacturer's catalogues, etc., is prepared to ease the design and selection procedure. The modules were combined under a supervisor program.

Programs are written in QuickBASIC and colour graphics are sometimes utilised to communicate with the user. The entire system is implemented on an IBM compatible PC. The use of CAD package for each module is illustrated with examples.

Key Words: Computer Aided Design, Machine Elements

ÖZET

MAKİNA ELEMANLARININ BİLGİSAYAR DESTEKLİ TASARIMI

DEMİRHAN, Necdet
Yüksek Lisans Tezi, Makina Mühendisliği Bölümü
Tez yöneticisi: Prof. Dr. İ. Hüseyin FİLİZ
Ağustos 1996, 155 sayfa

Bu tezde, makina elemanlarının bilgisayar destekli tasarımı için bir yazılım paketi geliştirilmiştir. Bu çalışmada mil, civata, rulman, helisel yay ve V-kayışı gibi makina elemanları ele alınmıştır.

Geliştirilen tasarım paketi modüler bir yapıya sahiptir. Yukarıda bahsedilen her bir makina elemanı için etkileşimli bir bilgisayar destekli tasarım/seçim programı yazılmıştır. Malzeme özellikleri, mukavemet yoğunluk faktörü gibi tasarım parametreleri, standart tablo, grafikler ve üretici firmaların katalog değerleri kullanılarak, seçim ve tasarım prosedürleri için bir veritabanı oluşturulmuştur. Bütün modüller bir ana program altında birleştirilmiştir.

Programlar QuickBASIC programlama dili ile yazılmış ve kullanıcının işini kolaylaştırmak için bilgisayarların grafik özelliklerinden yer yer yararlanılmıştır. Paket IBM uyumlu bir kişisel bilgisayarda geliştirilmiştir. Bilgisayar destekli tasarım paketinin her bir modülünün kullanımı ve sonuçları örneklerle gösterilmiştir.

Anahtar Kelimeler: Bilgisayar Destekli Tasarım, Makina Elemanları

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NOMENCLATURE

SHAFTS

d	Inner Diameter (mm)
D	Outer Diameter (mm)
D_{eq}	Equivalent Diameter (mm)
F	Axial Load (N)
k_a	Surface Factor
k_b	Size Factor
k_c	Reliability Factor
k_d	Temperature Factor
k_e	Modifying Factor for Stress Concentration
K_f	Fatigue Strength Reduction Factor
k_f	Miscellaneous-effects Factor
M	Bending Moment (N.m)
n	Factor of Safety
n_{crt}	Factor of Safety at Critical Section
q	Notch Sensitivity
S_e	Endurance Limit of Mechanical Element (MPa)
$S_{e'}$	Endurance Limit of Rotating Test Specimen (MPa)
S_f	Fatigue Strength (MPa)
S_{uc}	Ultimate Compression Strength (MPa)
S_{ut}	Ultimate Tensile Strength (MPa)
S_{yc}	Yield Strength in Compression (Mpa)

S_{yt}	Yield Strength in Tension (Mpa)
$S_{1,2}$	Normal Stress in Principal Direction (Mpa)
T	Torque (N.m)
σ'	Von Mises Stress (MPa)
σ_a	Alternating Stress or Stress Amplitude (MPa)
σ_m	Mean Stress (MPa)
σ_{max}	Maximum Bending Stress (MPa)
τ_{max}	Maximum Shear Stress (MPa)
σ_{min}	Minimum Bending Stress (MPa)
$\sigma_{x,y}$	Stress in x, y Directions (MPa)
τ_{xy}	Shear Stress in xy Plane (MPa)

ROLLING ELEMENT BEARINGS

C_x	Basic Dynamic Load Capacity
F_a	Actual Axial Bearing Load (N)
F_r	Actual Radial Bearing Load (N)
L_h	Basic Rating Life in Operating Hours
L_s	Average Life
L_{10}	Life in One Million Revolutions
n	Life Exponent
N	Rotational Speed (rpm)
P	Equivalent Dynamic Bearing Load
P_s	Equivalent Static Bearing Load

X	Dynamic Radial Load Factor
X_0	Static Radial Load Factor
Y	Dynamic Axial Load Factor
Y_0	Static Axial Load Factor

JOURNAL BEARINGS

c	Radial Clearance (mm)
C_i	Increment Value of Bearing Clearance (mm)
C_n	Initial Value of Bearing Clearance (mm)
C_x	Final Value of Bearing Clearance (mm)
D	Bearing Diameter (mm)
f	Coefficient of Friction
H	Power Loss Due to Friction
H_i	Minimum Film Clearance Due to Surface Roughness (mm)
H_o	Minimum Film Thickness (mm)
L	Bearing Length (mm)
LD	L/D ratio
L_i	Increment Value of Bearing Clearance (mm)
L	Initial Value of Bearing Length (mm)
L_x	Final Value of Bearing Length (mm)
N	Journal Speed (RPM)
O_1	Surface Roughness of Journal (rms)
O_2	Surface Roughness of Bearing (rms)
OL	Type of Oil (SAE)

P	Minimum Pressure in Bearing (MPa)
P_m	Recommended Unit Load (MPa)
P_s	Calculated Unit Load (MPa)
Q	Flow Rate of Oil (mm^3/s)
Q_s	Side Flow Rate of Oil (mm^3/s)
r	Bearing Radius (mm)
S	Bearing Characteristic Number
T_1	Inlet Temperature of Oil (C)
T_2	Outlet Temperature of Oil (C)
W	Radial Load on Bearing (N)
μ	Absolute Viscosity (Pa.s)

BOLTED JOINTS

D	Bolt Diameter (mm)
D_h	Bolt Hole Diameter (mm)
D_w	Washer Diameter (mm)
H	Nut Height (mm)
K_b	Bolt Stiffness (mm)
$K_{1,2}$	Stiffness of Each Member (N/mm)
K_m	Equivalent Stiffness of Members (N/mm)
R	Radial Coordinate
Z	Axial Coordinate
A_b	Cross Sectional Area of Bolt (mm^2)
A_m	Equivalent Cross Sectional Area of the Members (mm^2)

$E_{1,2}$	Modulus of Elasticity of Individual Member (GPa)
E_b	Modulus of Elasticity of Bolt Material (GPa)
E_{eq}	Equivalent Modulus of Elasticity of the Material of the Members
$L_{1,2}$	Individual Thickness of Each Member (mm)
L	Grip Length of the Bolted Joint (mm)
F_i	Preload (N)
P	External Tensile Load (N)
F_b	Resultant Bolt Load (N)
F_m	Resultant Load On Members (N)
N	Number of Bolts
S_{ut}	Ultimate Tensile Strength (MPa)
S_e	Endurance Limit For Machine Element (MPa)
n_s	Factor of Safety
α	Cone Angle
μ	Poissons's Ratio of The Material

HELICAL COMPRESSION SPRINGS

C	Spring Index
D	Mean Diameter (mm)
d	Wire Diameter (mm)
D_o	Outer Diameter (mm)
F_{rq}	Critical Frequency (cycle/second)
G	Modulus of Rigidity (GPa)

k	Spring Constant (N/mm)
L_f	Free Length (mm)
L_s	Solid Length (mm)
n	Factor of Safety
N_a	Number of Active Coils
N_d	Number of Dead Coils
N_t	Number of Total Coils
S_{se}	Shear Endurance Limit (MPa)
S_{sf}	Fatigue Shear Strength (Mpa)
S_{sy}	Shear Yield Strength (MPa)
S_y	Yield Strength (MPa)
τ	Static Shear Stress (MPa)
τ_a	Alternating Shear Stress (Mpa)
τ_m	Mean Shear Stress (Mpa)

V-BELT DRIVE SYSTEM

A	Belt Cross-section Area (mm)
B_l	Belt Life (hour)
B_n	Number of Belt
B_{pm}	Belt Passes per Minute
Bs$	Type of Belt Cross-section
C_d	Centre Distance (mm)
d	Small Shave Diameter (mm)

d_i	Driven Pulley Diameter (mm)
D	Large Shave Diameter (mm)
D_i	Driver Pulley Diameter (mm)
E	Modulus of Elasticity (MPa)
F_1	Tight-side Tension (N)
F_2	Slack-side Tension (N)
F_{b1}	Bending Tension on Small Shave (N)
F_{b2}	Bending Tension on Large Shave (N)
F_c	Centrifugal Force (N)
F_r	Radial Force (N)
F_s	Required Static Pretension (N)
F_t	Total Tension Force (N)
F_T	Total Peak Force (N)
F_{q1}	Flapping Frequency on Tight-side (Hz)
F_{q2}	Flapping Frequency on Slack-side (Hz)
F_{qn}	System Natural Frequency (Hz)
I_{T1}	Total Inertia of Driven Part (kg.mm^2)
I_{T2}	Total Inertia of Driver Part (kg.mm^2)
K_1	Arc Correction Factor
K_2	Belt Length Correction Factor
K_A	Speed Ratio Factor
K_T	Equivalent Torsional Stiffness
L_f	Free Length of Belt (mm)
L_p	Pitch Length of Belt (mm)
m	Mass of Belt per Length (kg/mm)

n_f	Speed of Faster Shave (rpm)
n_s	Speed of Slower Shave (rpm)
n_i	Speed of Driver Shaft (rpm)
n_o	Speed of Driven Shaft (rpm)
N_1	Force Peaks on Small Shave
N_2	Force Peaks on Large Shave
N_E	Equivalent Force Peaks
P	Driving Motor Power (Hp)
P_c	Corrected Rated Power (Hp)
P_d	Design Power (Hp)
P_r	Rated Power (Hp)
S_f	Service Factor
S_r	Speed Ratio
V	Belt Linear Velocity (m/s)
θ	Contact Angle on Small Pulley

CHAPTER I

INTRODUCTION

Mechanical engineering is considered as a branch of engineering which is primarily concerned with the design of machines to accomplish specific purposes. Machines are generally composed of several different mechanical elements which are designed to work together as a whole machine. Therefore, the efficiency of a machine depends strictly upon the design of individual machine elements.

The term 'design' can briefly be defined as finding suitable dimensions of the machine elements. Thus, each machine element should be designed such that it will perform its function without any failure. The design process is generally started with input data such as forces, power consumed or details like the tolerances or surface conditions. Some assumptions can also be made in order to simplify the work and to obtain a more compact formula to satisfy the requirements. The design is accomplished based on a selected design criteria. The principles of strength of materials, stress analysis, fatigue considerations, etc. are the most common keywords to the design. The objective is, however, to design a machine element that is not only sufficiently strong but also is cheap enough to be economically feasible. Therefore, sometimes the machine elements is better to select from manufacturer's catalogue than to design it in the laboratory. For example, the production or design of a rolling element bearing is a complicated and extra-high precision work. So, they are always selected among the bearings which are rated and standardised in the manufacturer's catalogues, as in the case of this work.

Increasing demands on the accuracy and lower cost, and the advent of CNC technology in industry and rapid progress in computer technology, required the developments of new techniques and methods in the design of the machine elements. Mechanical requirements have become equally important as the cost. The characteristics of the machine elements have significant influence on the precision of workpiece and productivity of a designed machine. Shafts, gears, bearings, springs, cams, belts, clutches, and brakes are common examples to the machine elements. These elements have received considerable amount of attention individually and well developed design and/or selection methods are available for these components in the literature. There is limited amount of works that involves a computer aided design package for the design and/or selection of machine elements. This work in the first place attempts to fill this gap with an interactive CAD package for the machine elements and to provide a tool for the designer to decide on the most suitable design among many alternatives. This work did not deal with the broader aspects of the design of machines, but attempt to accomplish the design of the separate elements which compose the machine. The developed interactive design and/or selection package consists of modules prepared for the design of each element, namely;

- shafts,
- anti-friction bearings,
- journal bearings,
- bolts,
- helical springs,
- V-belts,
- brakes.

The developed software has a modular structure. For each of the machine elements specified above, an interactive computer aided design/selection program is written. A database is constructed for the design and selection procedures by using corresponding standard tables, catalogue values and charts, etc. The modules were combined under a supervisor program.

Programs are written in QuickBASIC and colour graphics are sometimes utilised to communicate with the user. The entire system is implemented on an IBM compatible PC. The use of developed CAD package for each element is illustrated with examples. Many advantages are achieved from the use of developed CAD package including faster, better and cheaper design of machine elements.

Seven groups of machine elements are treated in this thesis. Since there are so many design parameters for each of those elements, list of symbols for each element is given seperately.

This thesis is organised as follows; Chapter 2 is devoted to computer aided design of shafts. In Chapter 3 the selection procedures for rolling element bearings (anti-friction bearings) are discussed, and computer aided selection of bearings are presented. Chapter 4 gives a general review of journal bearings and also involves the design of them with examples. Chapter 5 introduces the methodology used in the design of bolted joints by giving examples through the developed package. The theory and the design of helical springs are given in Chapter 6. V-belts and Brakes are handled in Chapter 7 and Chapter 8, respectively. Finally, Chapter 9 discusses the capability of the developed CAD package for the machine elements and the results obtained from this study.

CHAPTER II

SHAFTS

2.1. INTRODUCTION

In this chapter, general design principles for shafts are discussed. Failure theories for static and dynamic loading are briefed. Computer aided design of shafts is presented and the use of program is illustrated with certain examples.

2.2. CONSIDERATIONS ON THE DESIGN OF SHAFTS

Shafts are machine elements which transmit torque and support rotating machine components. Since the transmission of torque is associated with applied forces, such as forces acting on the teeth of gears, shafts are usually subjected to forces and bending moments in addition to the torque. Most shafts are subjected to combined bending and torsion, either of which may be steady or variable.

Shafts can be classified with respect to their purpose, as transmission shafts that carry drive members, such as gears, pulleys, chain sprockets, and clutches, and as main shafts which carry the operating members of machines, such as turbine wheels and disks, cranks, in addition to drive members.

With respect to their shape, shafts can be classified as uniform and stepped shafts. The diameter of a shaft along its length is determined by the load distribution and conditions imposed by the manufacturing and assembly processes used. Stepped shafts which may have equal-strength along its length are more preferable than uniform shafts.

Shafts are usually cylindrical in shape. They may be solid or hollow in design. The weight of a hollow shaft with a hole to outside diameter ratio of 0.75 is about half of a solid shaft of equal strength and rigidity.

Shafts may be rotating or stationary. If the load is steady and the shaft is stationary then the static stresses occur on the shaft. If the load is variable or the shaft is rotating, alternating and mean stresses occur.

There are many studies on the design of shafts and general design principles can be found in any textbook [1-5]. There are also many attempts to automate the design of shafts [6-20]. In this study, design principles are discussed and the algorithm for the automation of the design is presented. As different from previous works, a data base for materials, design parameters such as stress concentration factors, reliability factors, notch sensitivity factors, etc., are included and therefore a user could use this program without any difficulty.

Design of shafts requires a theory of failure to express a stress in terms of loads and shaft dimensions, and an allowable stress as fixed by material strength and safety factor. Maximum shear stress theory of failure and distortion energy theory of failure are the two most commonly used in shaft design.

2.3. DESIGN FOR STATIC STRENGTH

If a shaft or a beam is stationary and the load is of static type, only static type of stress are developed and the design is to be based on static strength and the procedure is dependent on the type of material as illustrated in Figure 2.1.

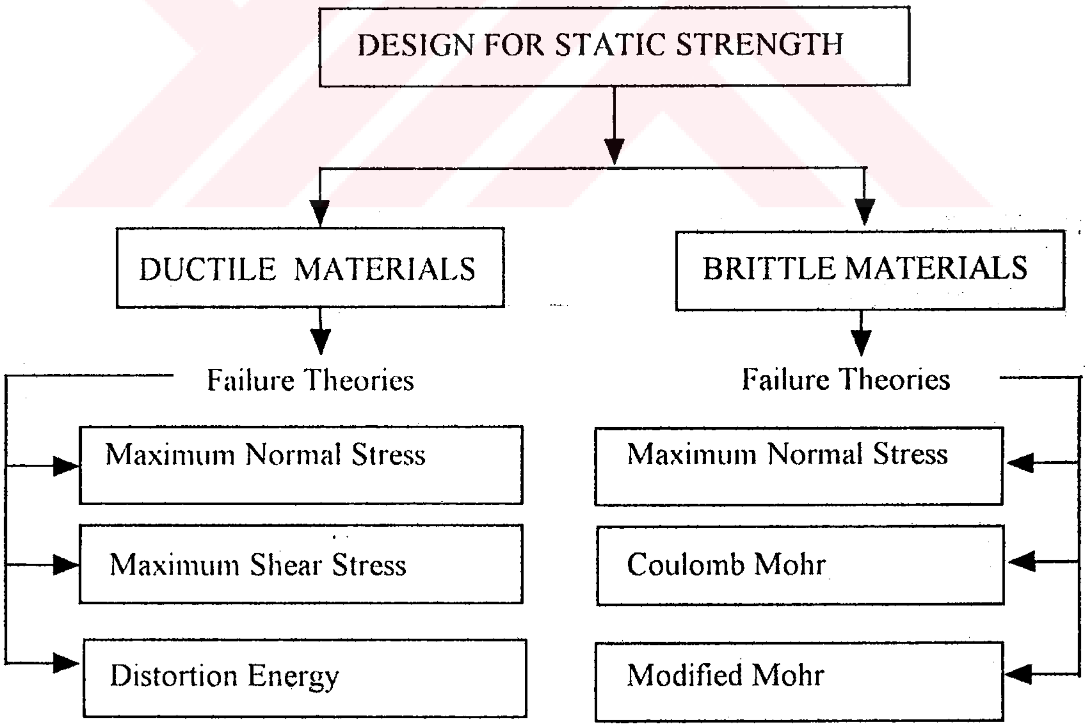


Figure 2.1 The Procedure for the Design for Static Strength

Failure theories used for the elements made of ductile materials are represented in Figure 2.2.

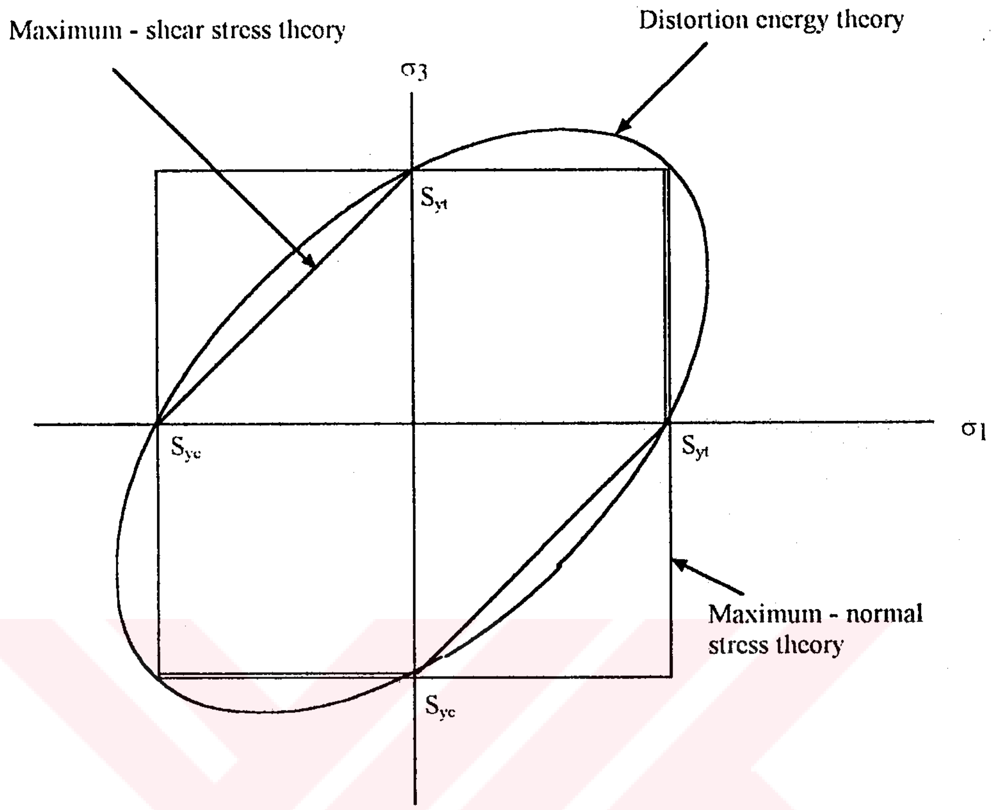


Figure 2.2 Graphical Representation of Failure Theories

According to maximum normal stress theory, failure will occur when the maximum normal stress in the element is equal to the normal stress in a tensile test specimen at the yield point or at fracture point. Maximum shear stress theory states that failure will occur when the maximum shear stress in the element is equal to the maximum shear stress in a tensile test specimen at the yield point. Distortion energy theory states that an element will fail if the distortion energy in that element is equal to the distortion energy in tensile test specimen at the yield point. Since yielding is considered as the failure criterion, maximum shear stress and distortion energy theories are applied only to ductile materials.

Among the above theories, maximum normal stress theory of failure is not recommended. Because it does not give correct results for the elements subjected to pure torsional loads.

In the most common loading which results biaxial stress state on the machine elements, principle stresses on the worstly loaded element are calculated as;

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \left[\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2 \right]^{1/2} \tag{2.1}$$

According to maximum-shear-stress theory, factor of safety n is determined from,

$$n = S_y / 2\tau_{max} \tag{2.2}$$

In the application of distortion-energy theory, Von Mises stress is needed to be calculated. Von Mises stress is:

$$\sigma' = S_1^2 + S_2^2 - S_1 S_2 \tag{2.3}$$

and factor of safety n is then calculated by,

$$n = \frac{S_y}{\sigma'} \tag{2.4}$$

If the shaft material is of brittle type and the loading is of static type, the following theories of failures are applicable;

a) Maximum Normal Stress theory of failure

b) The Coulomb Mohr theory of failure

c) The Modified Mohr theory of failure.

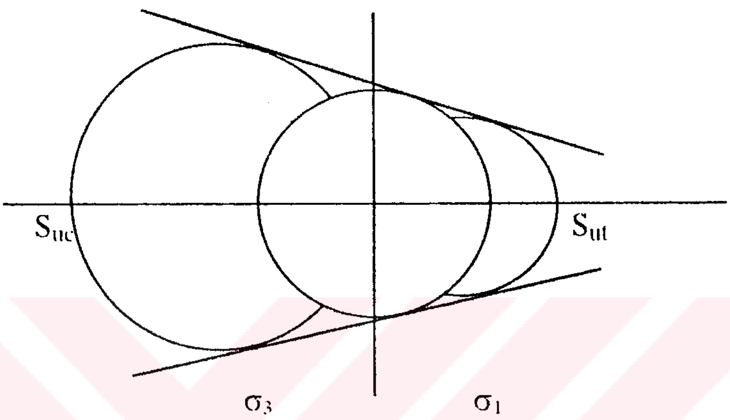


Figure 2.3 The Critical Stresses

The Coulomb Mohr theory states that failure will occur when the stress situation on an element results in a Mohr's circle which is tangent to the common tangent of the Mohr's circles when that element is loaded by a tensile and compression load separately.

The Modified Mohr theory is different from the Coulomb Mohr theory in the fourth quadrant in (σ_1, σ_3) plane. In the fourth quadrant, Modified Mohr theory is the same as maximum normal stress theory until σ_3 becomes less than $-S_{ut}$. These three theories of failure are represented in the following Figure 2.4.

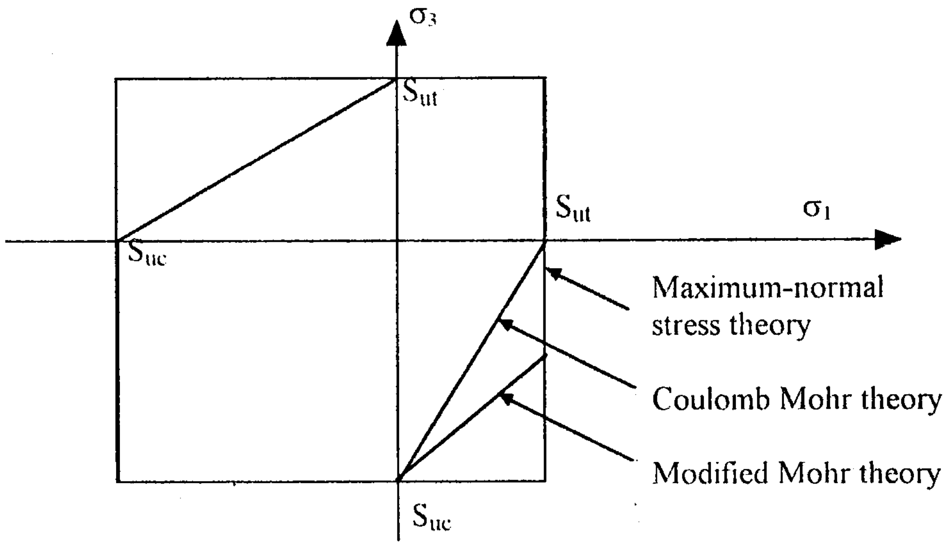


Figure 2.4 Three Failure Theories for Brittle Materials

Factor of safety is calculated as;

$$n = \frac{S_3}{\sigma_3} \tag{2.5}$$

where S_3 is the strength represented on the boundary of those failure theories and for Coulomb Mohr theory,

$$S_3 = \frac{S_{uc}}{\frac{S_{uc}}{\sigma_{ut}} \frac{\sigma_1}{\sigma_3} - 1} \tag{2.6}$$

and for Modified Mohr theory,

$$S_3 = \frac{S_{uc}}{\frac{S_{uc} - S_{ut}}{S_{ut}} \frac{\sigma_1}{\sigma_3} - 1} \tag{2.7}$$

In the design for static strength, stress concentration factors due to any discontinuity in the shaft cross-section are not taken into account. Only the stresses at the weakest section are calculated and then combined to determine principal stresses which are used in the above theories of failures.

2.4. DESIGN FOR FATIGUE STRENGTH

If the shaft is rotating, in addition to static stresses, reversed bending stresses and steady or repeating of torsional stresses may be developed. The most common type of loading is the load that will create reversed bending stresses and steady torsional stresses. Depending upon the type of elements mounted on the shaft, axial tension or compression stresses may also be developed and the designer must take these stresses into account.

In general fatigue design problems, the elements may be required to have infinite life or finite life. Infinite life requirement necessitates evaluation of endurance limit while life requirement needs determination of fatigue strength at that life. Fatigue strength corresponding to a specified life is calculated by using S-N diagrams. After finding the values of endurance limit or fatigue strength, the approaches described in the following figure might be used in designing or in analysing for fatigue strength. The relationship between life and fatigue strength is expressed as;

$$S_f = aN^b \qquad \text{where } a = \frac{(0.8S_{ut})^2}{S_e} \quad \text{and} \quad b = -\frac{1}{3}\log\frac{0.8S_{ut}}{S_e} \qquad (2.8)$$

In the case of the steels, after $N=10^6$ cycles, endurance limit S'_e starts and it is related to tensile strength. Experiments show that this relationship is;

$$S'_e = 0.5\, S_{ut}, \quad S_{ut} \leq 1400\ \text{MPa}$$

$$S'_e = 700\ \text{MPa}, \quad S_{ut} \geq 1400\ \text{MPa} \tag{2.9}$$

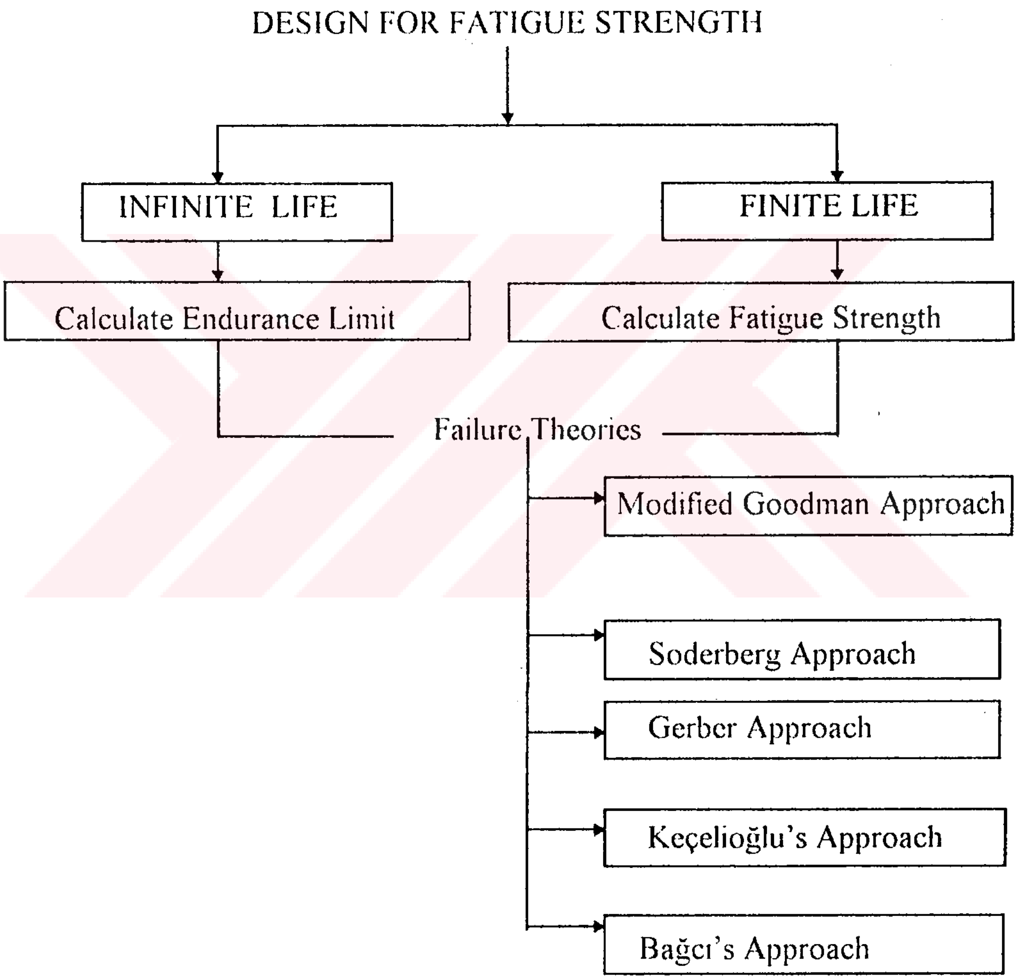


Figure 2.5 The Approaches for Fatigue Design

The Modified Goodman approach is a very conservative theory of the effect of mean stress on the fatigue resistance. The last three of the above approaches are non-linear theories and they appear to provide better job in predicting fatigue failure. It is probably not worth going extra trouble to use such theories unless all the strengths are known with some accuracy. In this thesis, we stick to the first two approaches as illustrated in Figure 2.6.

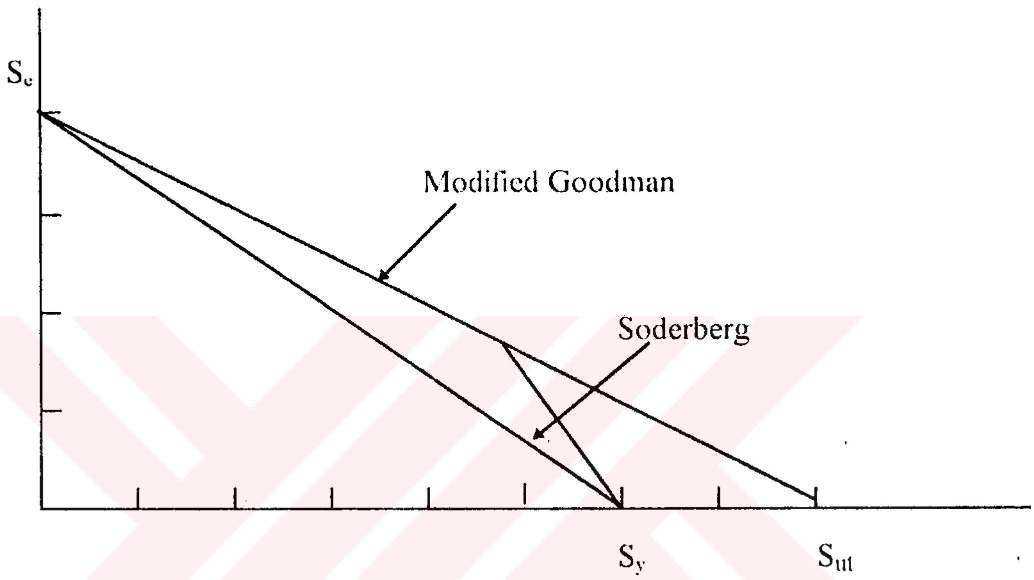


Figure 2.6 The Representation of Two Common Approaches

In the fatigue analysis, stresses at any point have alternating and mean components and they are defined as

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \qquad \text{and} \qquad \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \tag{2.10}$$

where σ_m is the midrange or mean stress, σ_a is the stress amplitude or alternating stress, σ_{max} is the maximum stress and σ_{min} is the minimum stress. Alternating and mean stress components are related to each other as follows. In Soderberg approach;

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = \frac{1}{n} \qquad (2.11)$$

Similarly, the Modified Goodman relation is found to be;

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n} \qquad (2.12)$$

In the calculation of endurance limit of the element, endurance limit modifying factors must be taken into account. These factors are :

a) k_a : Surface finish factor

b) k_b : Size factor

c) k_c : Reliability factor

d) k_d : Temperature factor

e) k_e : Modifying factor for stress concentration

f) k_f : Miscellaneous effects factor

Explanation of these factors can be found in any text book and also given in Appendix 1.

2.5. COMPUTER AIDED DESIGN OF SHAFTS

In this study, the shafts which carry two bearings and one or more forces i.e. gear forces, pulley forces and cutting forces are designed. The shaft may be a stepped or uniform shaft. The uniform shaft may have one or more discontinuities such as holes and grooves, but the stepped shafts may have key ways in addition to holes, grooves and shoulders.

Bearings are the supporting elements of the shaft. The forces on bearings, gears, and pulleys are assumed as concentrated loads which act at the mid point of the step. In one step, a shaft can only have one of the following; gear, pulley, cutting tool, hole, groove or key way. If there are two or more of these elements in one step, the step must be divided into two or more steps having same diameter. In the case of hollow shafts, the inner diameter may be uniform or stepped. Since the modelling of stepped hole is very difficult, the outer steps may be renumbered due to inner steps i.e. dividing the one step into two or more steps having same diameter. If the step has one of the bearing, gear or pulley, division must be made carefully, because load on these elements are assumed to affect on the middle of the step.

2.5.1 Computer Program

A general model for the shafts considered in this study is given in Figure 2.7. As illustrated in Figure 2.7, shafts may have complex structures whose design study would be laborious and time consuming.

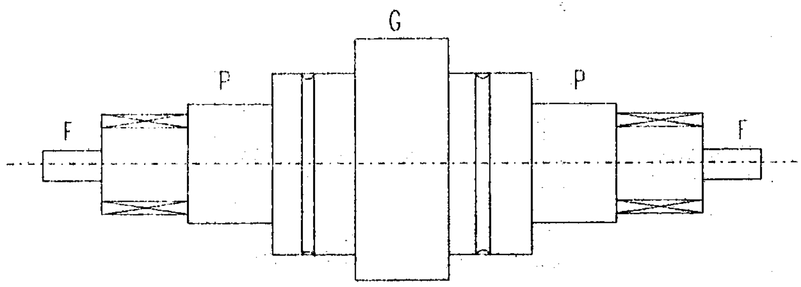


Figure 2.7 General Model for Shafts

The problems on the design of study of shafts have been categorised into three groups. They are;

1) All parameters may be specified, factor of safety is to be calculated.

2) Lengths of each step, material and factor of safety may be specified, diameters of each step are to be calculated.

3) All geometrical parameters and factor of safety may be specified, a suitable material is to be selected.

Factor of safety is calculated at five selected points. The first and fifth specified points are the starting and end point of the stepped shaft. The second, third and fourth points are the points either just before the force or discontinuity elements, at these elements, and just after these elements. If there is no force or discontinuity element on the step, then, second, third and fourth points are coincident at the midpoint of the step. The minimum of the calculated factor of safeties determines the critical section.

In case 1, the program is terminated when the factor of safety at the critical section is determined.

In case 2, starting from an initial diameter that is slightly greater than inner diameter, the factor of safety at the critical section is calculated and then compared with the given factor of safety. The diameter is increased until, the calculated factor of safety is greater or equal to the specified factor of safety.

In case 3, the material which has the lowest yield strength is first chosen. The factor of safety at critical section is then found and compared with the specified factor of safety. If the value found by the program is greater or equal to the specified one, the program is terminated. If not, the subsequent material from the database is selected and the procedure is continued until a satisfactory result is obtained. A simplified flowchart of the program is given in Figure 2.8.

When the program is executed, input data according to one of those alternatives mentioned above are set-up. Reaction forces at the supporting points are calculated and distributions of bending moments and torque along the length of the shaft are evaluated. After the evaluation of stress distribution, the worstly stressed element and the critical section is found. In strength calculations, if needed, relevant parameters from design database are selected by the program itself. With the database, the user is relieved from many monotonous actions like looking at tables and finding the suitable parameters.

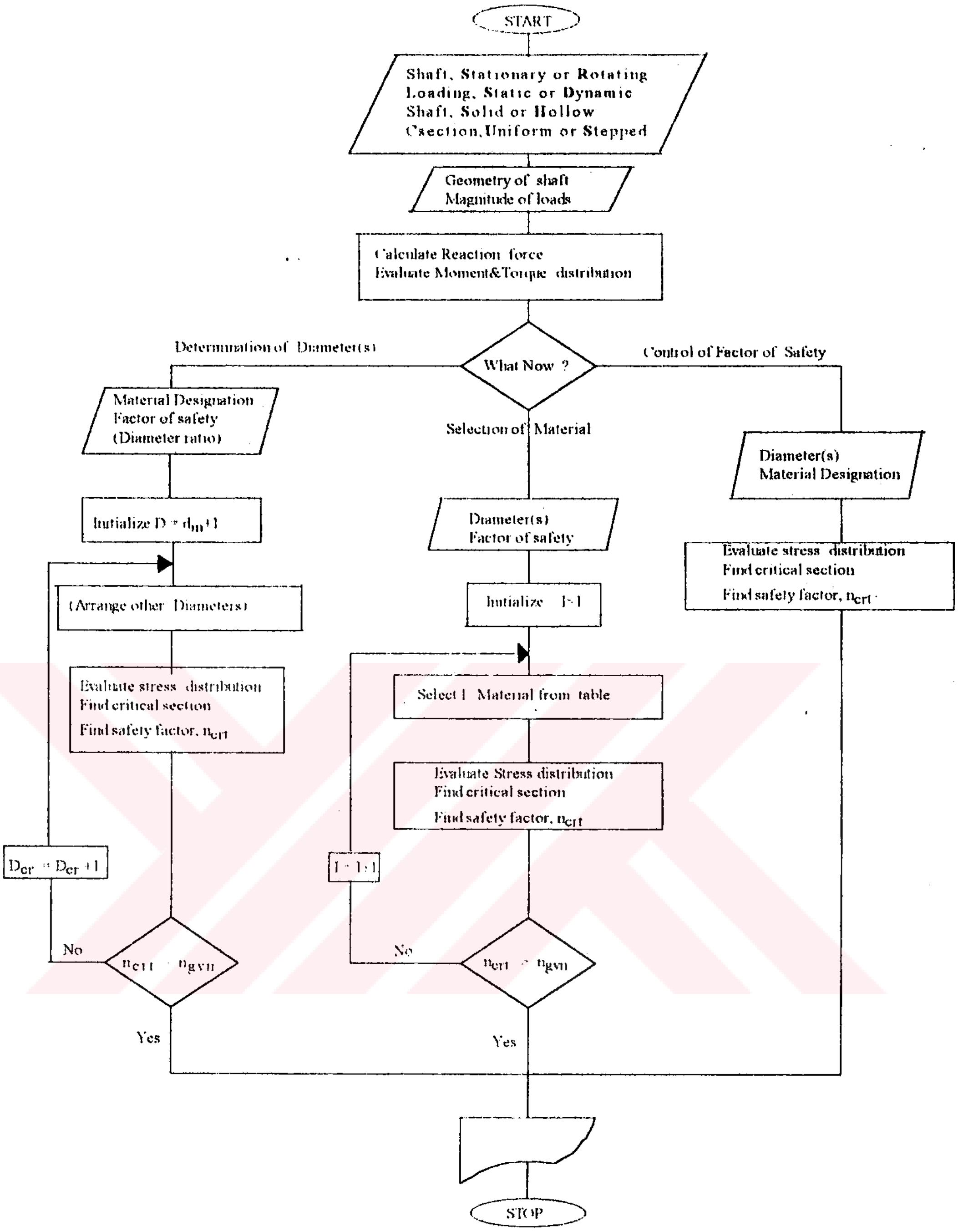


Figure 2.8 A Simplified Flowchart of the Program of Shaft Design

İNGİLİZCE ABSTRAKT (en fazla 250 sözcük) :

A computer aided design (CAD) package for the machine elements is prepared. Shafts, bolts, anti-friction bearings, journal bearings, helical springs, brakes and V-belts are the elements considered in this study.

The developed design package has a modular structure. For each of the machine elements specified above, an interactive computer aided design/selection program is written. A database which consists of material properties, design parameters like stress concentration factors, charts, standard values from manufacturer's catalogues, etc., is prepared to ease the design and selection procedure. The modules were combined under a supervisor program.

Programs are written in QuickBASIC and colour graphics are sometimes utilised to communicate with the user. The entire system is implemented on an IBM compatible PC. The use of CAD package for each module is illustrated with examples.

TÜRKÇE ABSTRAKT (en fazla 250 sözcük) :

(TÜBİTAK/TÜRDOK'un Abstrakt Hazırlama Kılavuzunu kullanınız.)

Makina elemanlarının bilgisayar destekli tasarımı için bir yazılım paketi geliştirilmiştir. Bu çalışmada mil, civata, rulman, helisel yay ve V-kayışı gibi makina elemanları ele alınmıştır.

Geliştirilen tasarım paketi modüler bir yapıya sahiptir. Yukarıda bahsedilen her bir makina elemanı için etkileşimli bir bilgisayar destekli tasarım/seçim programı yazılmıştır. Malzeme özellikleri, mukavemet yoğunluk faktörü gibi tasarım parametreleri, standart tablo, grafikler ve üretici firmaların katalog değerleri kullanılarak, seçim ve tasarım prosedürleri için bir veritabanı oluşturulmuştur. Bütün modüller bir ana program altında birleştirilmiştir.

Programlar QuickBASIC programlama dili ile yazılmış ve kullanıcının işini kolaylaştırmak için bilgisayarların grafik özelliklerinden yer yer yararlanılmıştır. Paket IBM uyumlu bir kişisel bilgisayarda geliştirilmiştir. Bilgisayar destekli tasarım paketinin her bir modülünün kullanımı ve sonuçları örneklerle gösterilmiştir.

2.5.2 Examples

The use of the program is illustrated with three examples for each case stated above. Due to space limitations, it was not possible to show all capability of the program for other alternatives.

Example 1: A high speed shaft of a worm-gear speed reducer made of UNS G61500 with S_y = 910 MPa and S_u = 1070 MPa, is subjected to a torque of 2100 Nm applied to the right end with no bending. The force on the worm has three components: a horizontal force, opposing rotation, of 28 kN, a vertical radial force of 8.8 kN, and a rightward thrust force of 30 kN. Dimensions are given. The left hand bearing takes the thrust load. All surfaces of the shaft are finished by grinding. Determine factor of safety on the possibility of fatigue failure for infinite life at 99% reliability.

SHAFT DESIGN

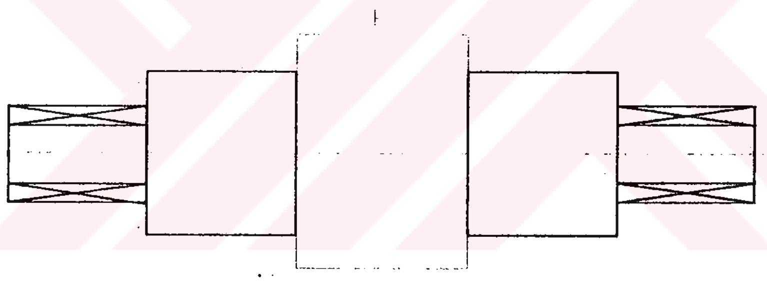
Shaft	: Stationary or Rotating	: Rotating
	Direction of rotation : Ccw or cW	: Cw
Life	: Infinite or Finite	: Infinite
Shaft	: Solid or Hollow	: Solid
Cross section	: Uniform or Stepped	: Stepped
	Number of Steps (NS >= 3)	: 5
	Step Number of Bearing- 1	: 1
	Step Number of Bearing- 2	: 5
	Which bearing carries thrust loads	: Right or Left : L
	Any discontinuities other than step	: Yes or No : N
Force type	: Transmitted or Single or Both	: Single
	Number of Forces < Max : 3 >	: 1
	Step Number of Force- 1	: 3
	Is any torque apply at the both end	: Yes or No : N
Design	: Diameter or Material or Control	: Control

Surface finish : Mach-cold drawn/ Hot rolled/ As forged/ Ground/ Polished : G
Reliability : 1>0.50 2>0.90 3>0.95 4>.99 5>.999 6>.9999 7>.99999 : 4
Operating temperature : Less or equal 350 C or Between 350 and 500 C : L

Desig.	Prcs.	Sy	Sut	HBN	Desig.	Prcs.	Sy	Sut	HBN
C1020	N	265	410	140A	C3315	H	640	885	220A
C1020	H	295	490	140A	C3330	H	540	735	230A
C1030	H	355	590	155A	C3415	H	885	1175	240B
C1030	N	235	440	155A	C3915	H	215	490	180A
C1030	H&T	295	540	155A	C4130	H&T	440	685	217C
C1040	H&T	325	590	172A	C4140	H	540	785	217A
C1040	N	275	540	172A	C5140	K	685	930	277A
C1050	N	335	590	260B	C5140	H&T	540	785	277A
C1050	H&T	355	590	260B	C5142	H&T	590	735	220B
C1060	N	385	685	243C	C8620	H	640	745	241C
C1060	H&T	430	685	243C	C8620	N	395	685	156D
C1090	H&T	885	1130	220A	C8640	H&T	905	1010	321D
C1117	A	215	375	0B	C8640	N	590	775	222A
C1350	H&T	980	1175	260D	C9245	H&T	1080	1275	255A
C3130	H&T	540	735	217A	C9255	H&T	1080	1275	290B
C3230	H&T	540	685	220A	C9265	H&T	1080	1275	310C

Use direction keys, select material, press Enter
If designation is different above, press <D>

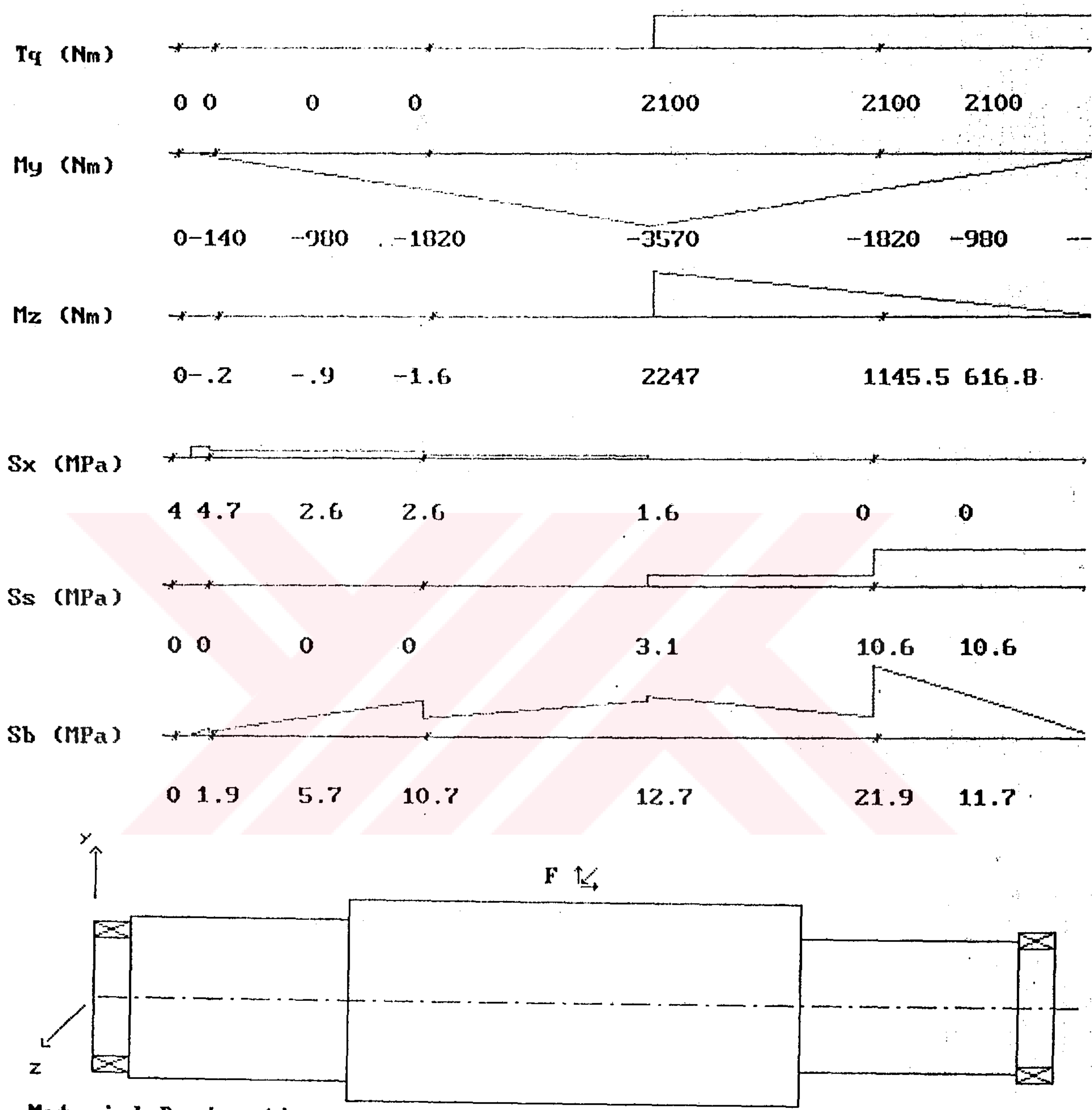
Designation : Sut = 1070 Sy = 910



All fillet radius are Equal or Not : N
All dimensions are in mm. All are numbered from left to right.
Step-1> Step length : 20 Outer diameter : 90 Fillet Radius : 3
Step-2> Step length : 120 Outer diameter : 120 Fillet Radius : 15
Step-3> Step length : 250 Outer diameter : 150 Fillet Radius : 15
X Coordinate of Point force-1 : 265
Step-4> Step length : 120 Outer diameter : 100 Fillet Radius : 3
Step-5> Step length : 20 Outer diameter : 90
The values are Right or Not :R

Values of Point force-1 (kN) : Fx = 30 Fy = -8.8 Fz = 28
α, Represented in figure, (deg) : 90
The values are Right or Not :R

The equilibrium of the torque is not satisfied.
The required torque applies from Right or Left end : R




```
Material Designation :
Section Lengths (mm) : 20.0 120.0 250.0 120.0  20.0
Section Diameters(mm): 90.0 120.0 150.0 100.0  90.0
Critical Diameter    : 100.00       Distance from Left : 390.00
Fxmax :     0.0   Tqmax : 2100.0    Mymax :-1820.0    Mzmax : 1145.5

Sxm :    0.0   Sbm :    0.0   Ssm :   10.7   Smen :  18.5    Se : 196.2
Sxa :    0.0   Sba :   21.9   Ssa :    0.0   Salt :  21.9
Using Soderberg theory, factor of safety              :  7.58
Using Modified Goodman theory, factor of safety       :  7.76
```

Diameter	Length	Fx-kN	Trq	My	Mz-Nm	Salt	Smen	Sf1	Sf2
90.00	0.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
90.00	10.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
90.00	10.00	-30.0	0.0	0.0	0.0	0.0	4.7	192.97	226.90
90.00	10.00	-30.0	0.0	0.0	0.0	0.0	4.7	192.97	226.90
90.00	20.00	-30.0	0.0	-140.0	-0.1	2.0	4.7	50.90	52.98
120.00	20.00	-30.0	0.0	-140.0	-0.1	0.8	2.7	171.86	185.78
120.00	80.00	-30.0	0.0	-980.0	-0.8	5.8	2.7	43.03	43.85
120.00	80.00	-30.0	0.0	-980.0	-0.8	5.8	2.7	43.03	43.85
120.00	80.00	-30.0	0.0	-980.0	-0.8	5.8	2.7	43.03	43.85
120.00	140.00	-30.0	0.0	-1820.0	-1.5	10.7	2.7	16.82	16.94
150.00	140.00	-30.0	0.0	-1820.0	-1.5	5.5	1.7	46.26	46.87
150.00	265.00	-30.0	0.0	-3570.0	-3.0	10.8	1.7	24.63	24.80
150.00	265.00	0.0	2100.0	-3570.0	2247.0	12.7	5.5	19.30	19.64
150.00	265.00	0.0	2100.0	-3570.0	2247.0	12.7	5.5	19.30	19.64
150.00	390.00	0.0	2100.0	-1820.0	1145.5	6.5	5.5	34.05	35.13
100.00	390.00	0.0	2100.0	-1820.0	1145.5	21.9	18.5	7.58	7.76
100.00	450.00	0.0	2100.0	-980.0	616.8	11.8	18.5	16.36	17.22
100.00	450.00	0.0	2100.0	-980.0	616.8	11.8	18.5	16.36	17.22
100.00	450.00	0.0	2100.0	-980.0	616.8	11.8	18.5	16.36	17.22
100.00	510.00	0.0	2100.0	-140.0	88.1	1.7	18.5	38.19	43.22
90.00	510.00	0.0	2100.0	-140.0	88.1	2.3	25.4	22.79	25.19
90.00	520.00	0.0	2100.0	-0.0	0.0	0.0	25.4	35.81	42.11
90.00	520.00	0.0	2100.0	-0.0	0.0	0.0	25.4	35.81	42.11
90.00	520.00	0.0	2100.0	-0.0	0.0	0.0	25.4	35.81	42.11
90.00	530.00	0.0	2100.0	-0.0	0.0	0.0	25.4	35.81	42.11

Example 2: The stationary shaft is subjected to completely reversed bending force of 44.4 kN. Shaft material has S_y = 620 MPa and S_u = 830 MPa. It is to be machined and is to have a life of 90000 cycles, corresponding to 99% reliability. Find a safe diameter based on a factor of safety of 1.5.

SHAFT DESIGN

Shaft	: Stationary or Rotating	: Stationary
Loading	: Static or Dynamic	: Dynamic
Life	: Infinite or Finite	: Finite
	Number of cycles	: 90000
Shaft	: Solid or Hollow	: Solid
Cross section	: Uniform or Stepped	: Stepped
	Number of Steps (NS >= 3)	: 4
	Step Number of Bearing- 1	: 1
	Step Number of Bearing- 2	: 4
	Which bearing carries thrust loads : Right or Left	: L
	Any discontinuities other than step : Yes or No	: N
Force type	: Transmitted or Single or Both	: Single

Number of Forces < Max : 2 > : 1
Step Number of Force- 1 : 3
Is any torque apply at the both end : Yes or No : N

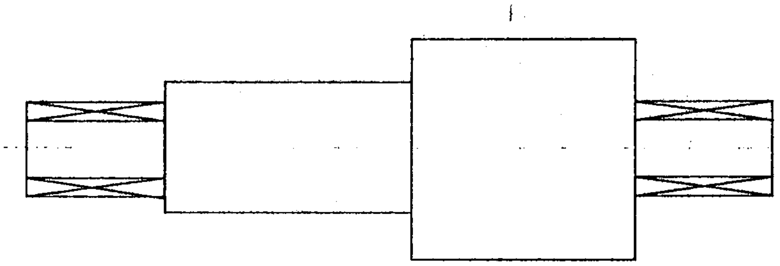
Design : Diameter or Material or Control : Diameter
Factor of Safety : 1.5

Surface finish : Mach-cold drawn/ Hot rolled/ As forged/ Ground/ Polished : M
Reliability : 1>0.50 2>0.90 3>0.95 4>.99 5>.999 6>.9999 7>.99999 : 4
Operating temperature : Less or equal 350 C or Between 350 and 500 C : L

Desig.	Prcs.	Sy	Sut	HBN	Desig.	Prcs.	Sy	Sut	HBN
C1020	N	265	410	140A	C3315	H	640	885	220A
C1020	H	295	490	140A	C3330	H	540	735	230A
C1030	H	355	590	155A	C3415	H	885	1175	240B
C1030	N	235	440	155A	C3915	H	215	490	180A
C1030	H&T	295	540	155A	C4130	H&T	440	685	217C
C1040	H&T	325	590	172A	C4140	H	540	785	217A
C1040	N	275	540	172A	C5140	K	685	930	277A
C1050	N	335	590	260B	C5140	H&T	540	785	277A
C1050	H&T	355	590	260B	C5142	H&T	590	735	220B
C1060	N	385	685	243C	C8620	H	640	745	241C
C1060	H&T	430	685	243C	C8620	N	395	685	156D
C1090	H&T	885	1130	220A	C8640	H&T	905	1010	321D
C1117	A	215	375	0B	C8640	N	590	775	222A
C1350	H&T	980	1175	260D	C9245	H&T	1080	1275	255A
C3130	H&T	540	735	217A	C9255	H&T	1080	1275	290B
C3230	H&T	540	685	220A	C9265	H&T	1080	1275	310C

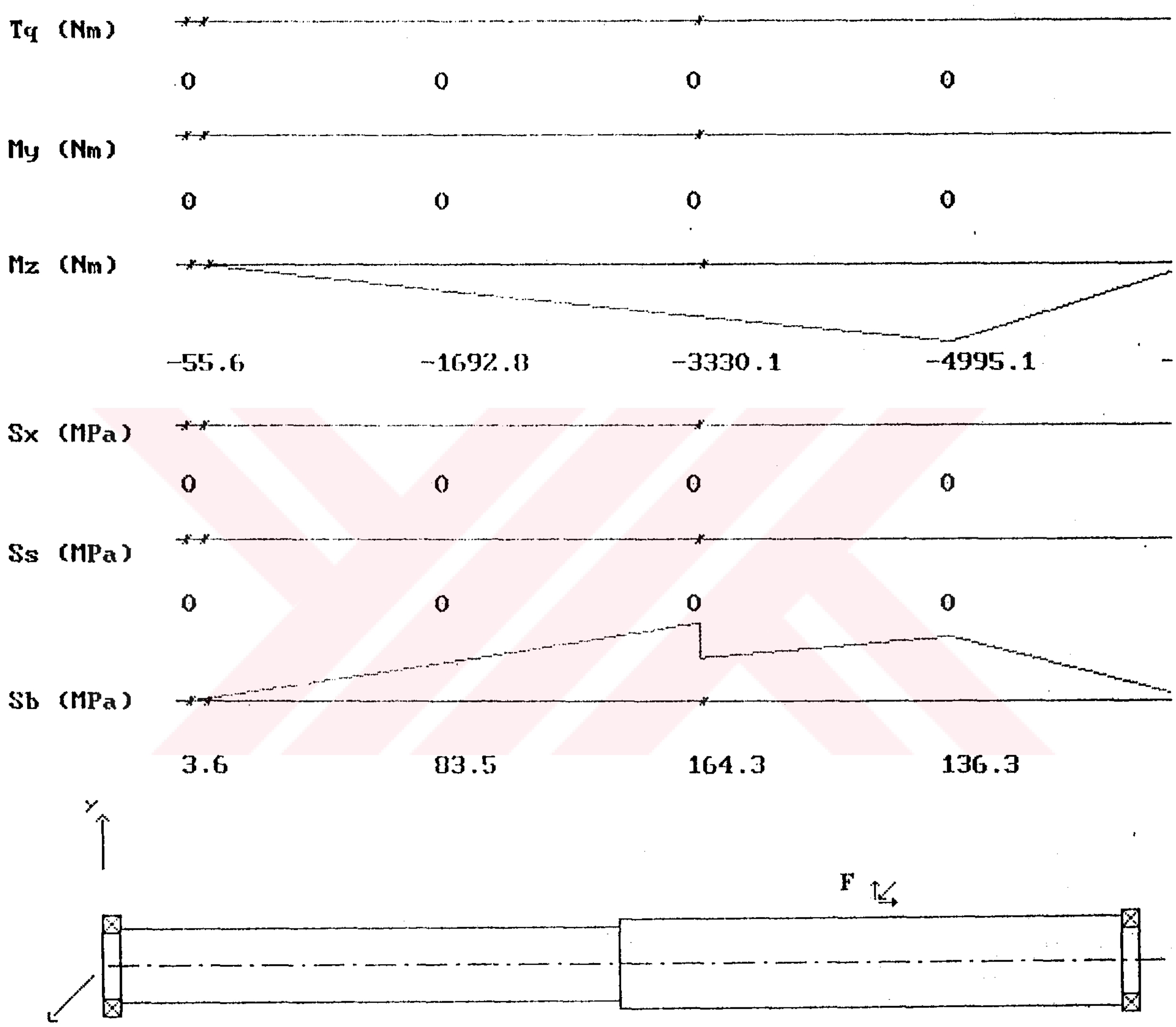
Use direction keys, select material, press Enter
If designation is different above, press <D>

Designation : Sut = 830 Sy = 620



All fillet radius are Equal or Not : E
Fillet Radius-mm : 6
All dimensions are in mm. All are numbered from left to right.
Step-1> Step length : 10
Step-2> Step length : 295 D2/D1 ratio : 1.1
Step-3> Step length : 295
X Coordinate of Point force-1 : 455
Diameter of Point of Application of Force-1 : 72
Step-4> Step length : 10 D4/D1 ratio : 1
The values are Right or Not :R

Point force-1 is Static or Dynamic : D
Max. values (kN) : Fx = 0   Fy = 44.4   Fz = 0
Min. values (kN) : Fx = 0   Fy = -44.4   Fz = 0
α, Represented in figure, (deg) : 90
The values are Right or Not : R




```
Material Designation :
Section Lengths (mm) : 10.0 295.0 295.0  10.0
Section Diameters(mm): 53.7  59.1  72.0  53.7
Critical Diameter     :  59.10      Distance from Left : 305.00
Fxmax :     0.0   Tqmax :     0.0   Mymax :     0.0   Mzmax :-3330.0
Fxmin :     0.0   Tqmin :     0.0   Mymin :     0.0   Mzmin : 3330.0
Sxm :   0.0   Sbm :   0.0   Ssm :   0.0   Smen :   0.0   Se : 147.5
Sxa :   0.0   Sba : 164.3   Ssa :   0.0   Salt : 164.3   Sf : 249.2
Using Soderberg theory, factor of safety          :  1.52
Using Modified Goodman theory, factor of safety   :  1.52
```

Diameter	Length	Fx-kN	Trq	My	Mz-Nm	Salt	Smen	Sf1	Sf2
53.73	0.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
53.73	5.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
53.73	5.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
53.73	5.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0:00
53.73	10.00	0.0	0.0	0.0	-55.5	3.6	0.0	70.30	70.30
59.10	10.00	0.0	0.0	0.0	-55.5	2.7	0.0	120.23	120.23
59.10	157.50	0.0	0.0	0.0	-1692.8	83.5	0.0	3.94	3.94
59.10	157.50	0.0	0.0	0.0	-1692.8	83.5	0.0	3.94	3.94
59.10	157.50	0.0	0.0	0.0	-1692.8	83.5	0.0	3.94	3.94
59.10	305.00	0.0	0.0	0.0	-3330.0	164.3	0.0	1.52	1.52
72.00	305.00	0.0	0.0	0.0	-3330.0	90.9	0.0	3.58	3.58
72.00	455.00	0.0	0.0	0.0	-4995.0	136.3	0.0	2.39	2.39
72.00	455.00	0.0	0.0	0.0	-4995.0	136.3	0.0	2.39	2.39
72.00	455.00	0.0	0.0	0.0	-4995.0	136.3	0.0	2.39	2.39
72.00	600.00	0.0	0.0	0.0	-166.5	4.5	0.0	71.56	71.56
53.73	600.00	0.0	0.0	0.0	-166.5	10.9	0.0	22.99	22.99
53.73	605.00	0.0	0.0	0.0	-0.0	0.0	0.0	4.1E6	4.1E6
53.73	605.00	0.0	0.0	0.0	-0.0	0.0	0.0	4.1E6	4.1E6
53.73	605.00	0.0	0.0	0.0	-0.0	0.0	0.0	4.1E6	4.1E6
53.73	610.00	0.0	0.0	0.0	-0.0	0.0	0.0	4.1E6	4.1E6

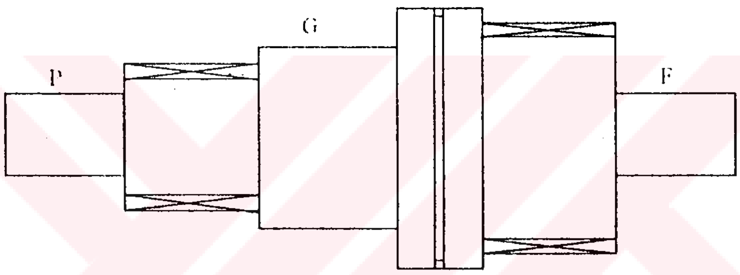
Example 3: A high speed shaft of a helical gear is subjected to torque by means of a pulley at one end. The pulley has a driven force of 8 kN and a driver force of 6 kN. The shaft is also loaded by a cutting force at other end which has three components; a horizontal force of -12 kN, a vertical radial force of -4 kN, and a tangential force of 6 kN. Helical gear which is placed at the mid section of the shaft, transmits a load of 10 kN. The shaft is to machined and is to have an infinite life, corresponding to 99 percent reliability. Select a suitable material for the shaft corresponding to a factor of safety of 1.5.

SHAFT DESIGN

Shaft	: Stationary or Rotating	: Rotating
	Direction of rotation : Ccw or cW	: Cw
Life	: Infinite or Finite	: Infinite
Shaft	: Solid or Hollow	: Solid
Cross section	: Uniform or Stepped	: Stepped
	Number of Steps (NS >= 3)	: 6
	Step Number of Bearing- 1	: 2

Step Number of Bearing- 2 : 5
Which bearing carries thrust loads : Right or Left : L
Any discontinuities other than step : Yes or No : Y
Is there any hole on a groove : Yes or No : N
Discontinuity type : Groove or Hole or Both : Groove
Number of Grooves < Max : 3 > : 1
Step Number of Groove- 1 : 4
Force type : Transmitted or Single or Both : Both
Transmission type : Gear or Pulley or Both : Both
Step Number of Gear- 1 : 3
Step Number of Pulley- 1 : 1
Is any torque apply at the both end : Yes or No : N
Design : Diameter or Material or Control : Material
Factor of Safety : 1.5

Surface finish : Mach-cold drawn/ Hot rolled/ As forged/ Ground/ Polished : M
Reliability :1>0.50 2>0.90 3>0.95 4>.99 5>.999 6>.9999 7>.99999 : 4
Operating temperature : Less or equal 350 C or Between 350 and 500 C : L

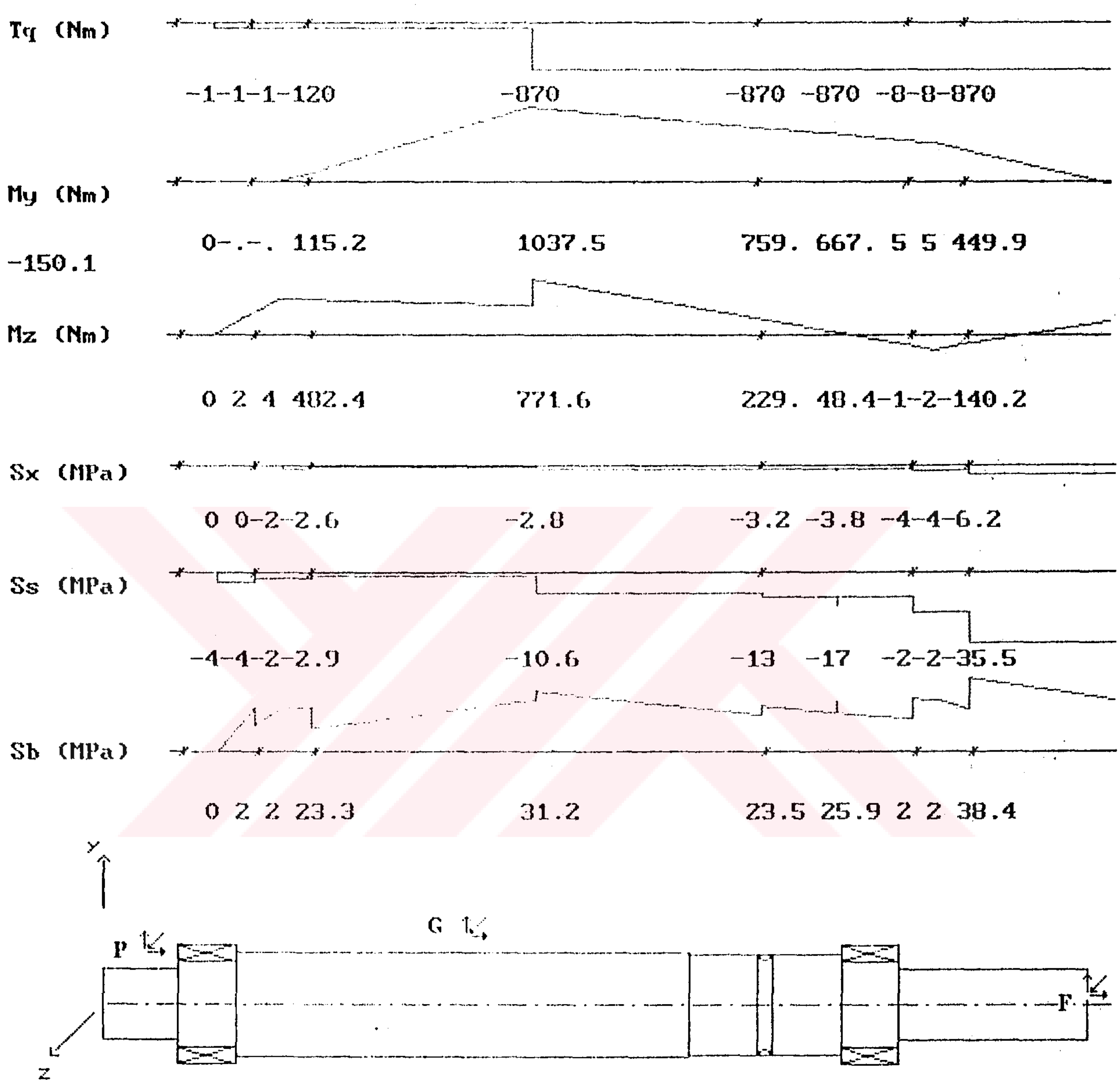


All fillet radius are Equal or Not : E
Fillet Radius-mm : 5
All dimensions are in mm. All are numbered from left to right.
Step-1> Step length : 40 Outer diameter : 50
Step-2> Step length : 30 Outer diameter : 60
Step-3> Step length : 240 Outer diameter : 75
Step-4> Step length : 80 Outer diameter : 70
Radius of Groove-1 : 3
X Coordinate of Groove-1 : 350
Step-5> Step length : 30 Outer diameter : 60
Step-6> Step length . 100 Outer diameter : 50
X Coordinate of Point force-1 : 520
The values are Right or Not :R

Magnitudes of Pulley force-1 (kN) : P1 = 8 P2 = 6
Output force must be > than input. Press any key to continue
Magnitudes of Pulley force-1 (kN) : P1 = 6 P2 = 8
α1, Represented in figure, (deg) : 90 α2 : 90
Diameter of Pulley-1 (mm) : 120
Mounting is interference Fit or Keyway : F
The values are Right or Not :R

The Transmitted load or Components of gear force-1 are known : T
Type of gear-1 is Spur, Helical, Worm or Bevel : H
Normal pressure angle : 20
Helix Angle : 25
Right or Left Hand Helix : R
Magnitude of Transmitted Load (kN) : 10
Gear is Driving or driveN : N
α, Represented in figure (deg) : 90
Pitch diameter : 150
Mounting is interference Fit or Keyway : K
Stress Concentration Factor of Keyway : 1.5
The values are Right or Not :R

Axial component of force-1 is Static or Dynamic : S
Values of Point force-1 (kN): Fx=-12 Fy=-4 Fz= 6
Diameter of point of application (mm) : 50
α, Represented in figure, (deg) : 60
The values are Right or Not :R



Material Designation : C1040
Section Lengths (mm) : 40.0 30.0 240.0 80.0 30.0 100.0
Section Diameters(mm): 50.0 60.0 75.0 70.0 60.0 50.0
Critical Diameter : 50.00 Distance from Left : 420.00
Fxmax : 12.0 Tqmax : -870.0 Mymax : 450.0 Mzmax : -140.2

Sxm : -6.1 Sbm : 0.0 Ssm : -35.4 Smen : 61.7 Se : 102.1
Sxa : 0.0 Sba : 38.4 Ssa : 0.0 Salt : 38.4
Using Soderberg theory, factor of safety : 1.67
Using Modified Goodman theory, factor of safety : 2.04

Diameter	Length	Fx-kN	Trq	My	Mz-Nm	Salt	Smen	Sf1	Sf2
50.00	0.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
50.00	20.00	0.0	0.0	0.0	0.0	0.0	0.0	0.00	0.00
50.00	20.00	0.0	-120.0	0.0	0.0	0.0	8.5	32.47	63.77
50.00	20.00	0.0	-120.0	0.0	0.0	0.0	8.5	32.47	63.77
50.00	40.00	0.0	-120.0	-0.0	280.0	22.8	8.5	3.93	4.18
60.00	40.00	0.0	-120.0	-0.0	280.0	13.2	4.9	9.42	10.27
60.00	55.00	0.0	-120.0	-0.0	490.0	23.1	4.9	5.80	6.11
60.00	55.00	7.3	-120.0	-0.0	490.0	23.1	5.5	5.72	6.07
60.00	55.00	7.3	-120.0	-0.0	490.0	23.1	5.5	5.72	6.07
60.00	70.00	7.3	-120.0	115.3	482.4	23.4	5.5	3.78	3.93
75.00	70.00	7.3	-120.0	115.3	482.4	12.0	3.0	7.48	7.79
75.00	190.00	7.3	-120.0	1037.6	421.9	27.0	3.0	3.47	3.54
75.00	190.00	12.0	-870.0	1037.6	771.7	31.2	18.4	2.58	2.82
75.00	190.00	12.0	-870.0	1037.6	771.7	31.2	18.4	2.58	2.82
75.00	310.00	12.0	-870.0	759.9	229.2	19.2	18.4	3.80	4.34
70.00	310.00	12.0	-870.0	759.9	229.2	23.6	22.6	3.03	3.45
70.00	350.00	12.0	-870.0	667.3	48.4	19.9	22.6	4.61	5.66
64.00	350.00	12.0	-870.0	667.3	48.4	26.0	29.5	2.25	2.56
70.00	350.00	12.0	-870.0	667.3	48.4	19.9	22.6	4.61	5.66
70.00	390.00	12.0	-870.0	574.7	-132.4	17.5	22.6	4.97	6.22
60.00	390.00	12.0	-870.0	574.7	-132.4	27.8	35.8	2.40	2.83
60.00	405.00	12.0	-870.0	540.0	-200.2	27.2	35.8	3.21	4.03
60.00	405.00	12.0	-870.0	540.0	-200.2	27.2	35.8	3.21	4.03
60.00	405.00	12.0	-870.0	540.0	-200.2	27.2	35.8	3.21	4.03
60.00	420.00	12.0	-870.0	450.0	-140.2	22.2	35.8	3.59	4.65
50.00	420.00	12.0	-870.0	450.0	-140.2	38.4	61.7	1.67	2.04
50.00	520.00	12.0	-870.0	-150.0	259.8	24.4	61.7	2.60	3.64
50.00	520.00	0.0	-790.1	-0.0	-0.0	0.0	55.8	4.93	9.68
50.00	520.00	0.0	-790.1	-0.0	-0.0	0.0	55.8	4.93	9.68
50.00	520.00	0.0	-790.1	-0.0	-0.0	0.0	55.8	4.93	9.68

CHAPTER III

SELECTION OF ROLLING ELEMENT BEARINGS

3.1. INTRODUCTION

In this chapter, computer aided selection of rolling element bearings for different loading conditions, is discussed

Selection procedure involves selection of bearing type and size by considering load, life and speed requirements. Type of bearings considered in this thesis are discussed in section 3.2. Section 3.3 is devoted to the discussion of selection criteria. In section 3.4, bearing analysis for different loading conditions is presented. Load-life relationship is explained in section 3.5. Computer aided selection of rolling element bearings is discussed and examples are presented in section 3.6.

3.2. TYPE OF BEARINGS

Rolling element bearings are classified into two main groups, namely ball bearings and roller bearings. Each group may be divided into two subgroups: radial bearings and thrust bearings. Since most types of radial bearings can carry thrust load, and some thrust bearings can carry radial load, there is no clear distinction between these two subgroups. However, one important difference is that for radial bearings the load carrying capacity is given as pure radial load, while for thrust bearings the load carrying capacity is given as pure thrust load [21]. The most common types of bearings are given below:

a) Ball bearings

1. Deep groove ball bearings
2. Self aligning ball bearings
3. Single row angular contact ball bearings
4. Double row angular contact ball bearings
5. Four point contact ball bearings
6. Thrust (single direction) ball bearings
7. Thrust (double directions) ball bearings.

b) Roller bearings

1. Cylindrical roller bearings
2. Needle roller bearings
3. Spherical roller bearings

4. Taper roller bearings

5. Cylindrical thrust roller bearings

6. Needle thrust roller bearings

7. Spherical thrust roller bearings

Each one of the above bearings has certain performance characteristics which determine its suitability for specific applications. Ball bearings are generally the best for light and medium loads, high speed and small diameter of the shaft. However, for large and heavily loaded applications roller bearings are preferred.

3.3. SELECTION CRITERIA OF BEARINGS

Because of the many different types of bearings that are available, type selection is not simple. It is rare that the features of a bearing fit exactly the specific requirements of an application. Consequently selection of bearing type is usually based on the factor considered most important. The various bearing types can be examined from standpoint of the most common selection criteria which is given in the following :

a) Loading condition

b) Rotational speed

c) Life

d) Mechanical requirements

- i) Stiffness
- ii) Inertia

iii) Friction and Lubrication

e) Space

f) Noise

g) Cost

In the selection procedure, the factors a), b), and c) come first. However, the bearing must be checked against stiffness, inertia and friction requirements. In some applications, such as spindles and feed drive systems of machine tools, stiffness is considered to be the finalising factor in the selection of the bearing types and sizes. Space requirements are not usually a serious consideration in most bearing applications, but can be important when space is to be minimised. The errors in the rolling elements and raceways are the main cause of the noise. Improving the quality of the rolling elements and raceways reduces the noise level. Cost is another factor but not very critical.

3.4 ANALYSIS OF BEARINGS

The basic function of the bearings is to carry the loads under a given operating conditions for the required life. Therefore, the first step in the selection of bearings is to evaluate the loads. These loads are transmitted through the rolling elements from the inner race to the outer. The magnitude of the loading carried by the individual ball or roller depends on the internal geometry of the bearing and on the type of load imposed on it. In addition to applied loading, rolling elements are also subjected to dynamic loading due to speed effect, but in the case of moderate speed of rotation, dynamic loading can be ignored.

The method of determination of the loads acting at the supporting points is dependent on the bearing configuration on the shafts.

In the standard design, a common bearing arrangement that separately takes up the radial and axial forces is shown in Figure 3.1. The radial loads are carried by double row cylindrical roller bearings and the axial loads are carried by double acting thrust ball bearings. Using double rows ensure a high bearing load-carrying capacity and rigidity. For large shaft diameters, the circumferential bearing speeds will be considerably higher, therefore instead of thrust ball bearings, double row angular contact thrust ball bearings are used. By using simple equilibrium equations, it is possible to determine axial and radial loads and proceed with the selection process.

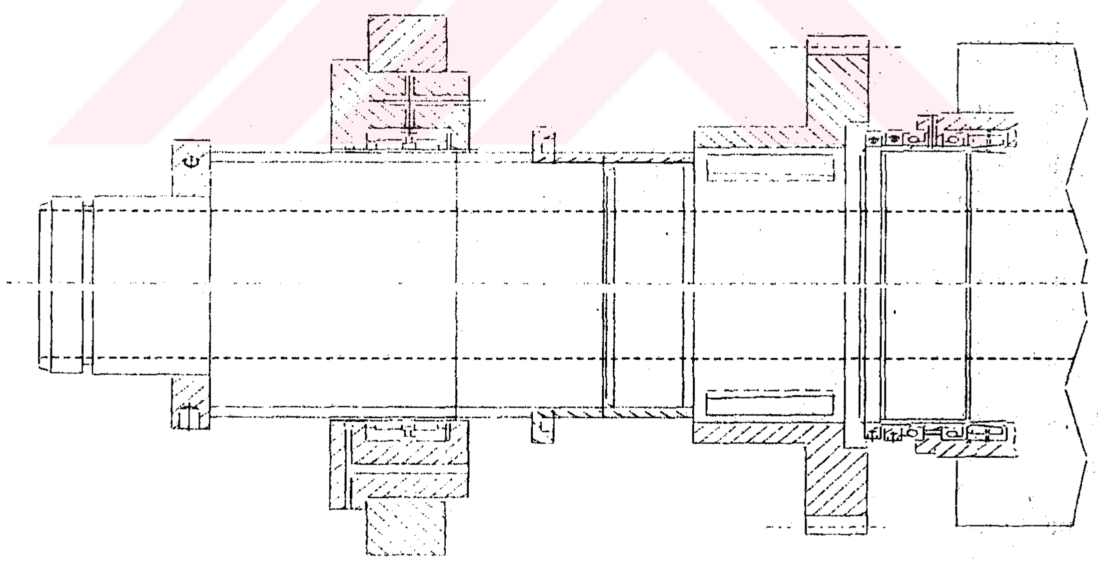


Figure 3.1 A Typical Bearing Arrangement [22]

In the bearings which are subjected to radial loads, elastic deformations develop at and adjacent to the point of contact. Radial bearings can be mounted on the shaft with a clearance, or with an interference (preload). Most of the time, radial preloading is necessary to increase radial stiffness of the bearings because it increases the number of rolling elements under load and thus reduces the maximum rolling element load. Radial preloading is obtained by making the shaft diameter greater than the bore diameter of the bearing at the required amount.

The advantages and disadvantages of the radial preloading are discussed by Harris [23], Swanson [24] and Leibensperger [25]. They stated that an excessive amount of preload reduces the life of the bearings. One must note that radial preloading will not effect the calculation of radial reactions but it affects the load distribution among the rolling elements and improves the performance of the bearings.

Another configuration is used to accommodate axial load by using angular contact thrust ball bearings and radial load by double row cylindrical roller bearings. This configuration is shown in Figure 3.2. The analysis is the same as in the previous configuration.

In some applications, the stiffness requirement is the dominating factor in selecting anti-friction bearings. The simple solution to this problem is the selection of preloaded taper roller or angular contact ball bearings. They may be used in arrangements of back-to-back, tandem, face-to-face or any combinations of these [26].

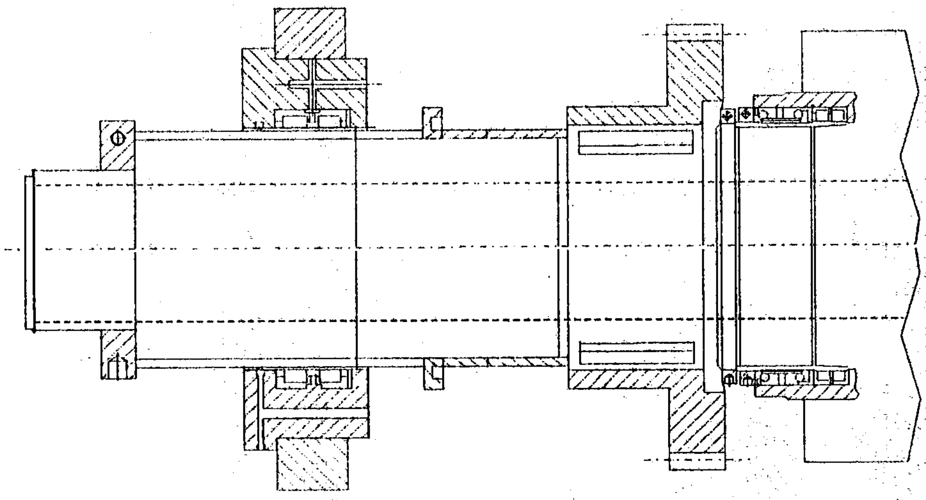


Figure 3.2 A Spindle Bearing Arrangement of a Finishing Lathe [22]

As shown in Figure 3.3, two single row taper roller bearings which have high axial stiffness and high axial load carrying capacity are mounted in back to back arrangement.

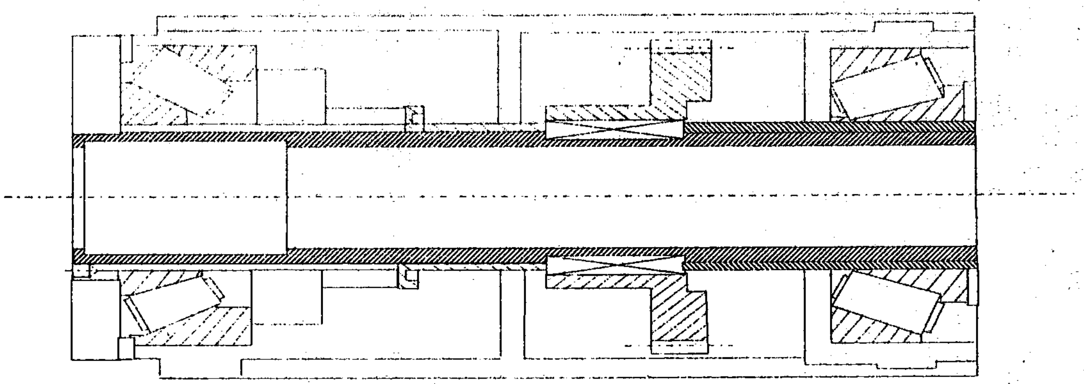


Figure 3.3 A typical Bearing Arrangement on a Shaft

In the above type of applications, bearings are subjected to combined loading and usually preloaded in axial directions. Evaluation of reactions at the supporting points is a cumbersome work since the force system will become indeterminate after preloading. In addition to force equilibrium equations compatibility conditions for deflections must be included as may be visualised from following figures.

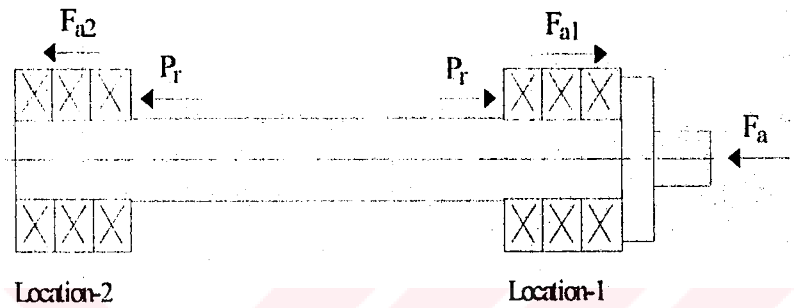


Figure 3.4 A Schematic Representation of an Axially Preloaded Bearing Assembly

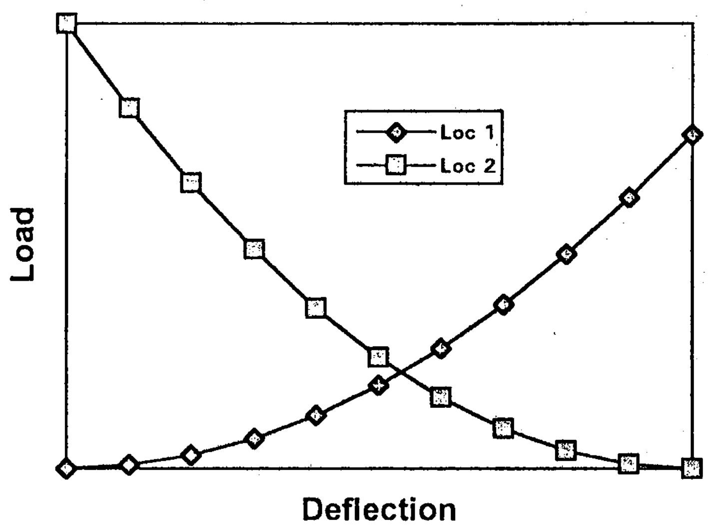


Figure 3.5 Load Deflection Diagram of Preloaded Ball Bearings

The analyses of bearings under axial loading and combined loading are performed by some researchers. Bearings under thrust loading have been analysed by Filiz and Bell [27]. Bearings under combined loading have been analysed by Filiz and Görür [28]. These analyses have left out of the scope of this thesis and their algorithms developed for automated calculations are adopted in the program prepared in this thesis. One may refer to these papers for more information.

3.5 LIFE - LOAD RELATIONSHIP IN BEARINGS

Bearing life, under a given loading condition, is defined as the number of revolutions or the number of hours at a given rotational speed at which the bearing will operate without fatigue developing.

There is an exponential relationship between the life and magnitude of the load and in general, it is expressed as the following;

$$L_{10} = \left(\frac{C_d}{P}\right)^n \tag{3.1}$$

where L_{10} is the life in 10^6 revolutions and n is the life exponent (n=3 for ball bearing, n=10/3 for roller bearings). In terms of hours, the life can be expressed as;

$$L_h = \frac{16667}{N}\left(\frac{C_d}{P}\right)^n \tag{3.2}$$

Basic dynamic load capacity C_d, is the load that the bearing can carry for one million revolutions before fatigue failure occurs. It depends on the size and number of the rolling elements, contact angle under the load, hardness of the material, properties of the lubricant, operating temperature and operating speed. P is the radial load for radial bearings and is the thrust load for thrust bearings and is the equivalent radial load for combined loading;

$$P = X F_r + Y F_a \tag{3.3}$$

and

$$P_o = X_o F_r + Y_o F_a \tag{3.4}$$

where X, X_o, Y, Y_o are axial and radial load factors and they are specified in manufacturers' catalogues. The load as well as the speed of the bearings may be constant or may be variable with time. For variable loads and speeds, mean values must be calculated whose formulae can be found in any text book or in any manufacturers' catalogue. Basic load dynamic capacities are automatically taken from the database and the life of the bearings are calculated for each type of bearing. If more than one bearing is mounted on the shaft, Harris [29] gave an empirical formula for the calculation of average life of the set which contains m number of bearings as:

$$L_s = \left(\sum_{i=1}^{m} L_i^{\frac{10}{9}} \right)^{\frac{9}{10}} \tag{3.5}$$

3.6 COMPUTER AIDED SELECTION OF ROLLING ELEMENT BEARINGS

3.6.1 Computer Program

The calculations involved in selecting a bearing or multiple bearings in engineering design applications are complicated and laborious. From among many types and sizes of the bearings, sometimes with conflicting requirements such as stiffness and inertia, selection is based on a weighted compromise between these requirements.

In the selection procedure, firstly, bearings are selected so as to satisfy the load carrying capacity and life requirements. The limiting speed of the bearing must also be checked against the spindle speed. If satisfactory, then deflection and stiffness for the given loading conditions, are calculated. The selection process is terminated if the stiffness is considered to be satisfactory.

As discussed in the previous sections, there are two basic bearing configurations in which the bearings may be subjected to pure radial load, pure axial load, or combination of axial and radial loads.

Considering the types of loads and configurations, the computer program is prepared on modular basis. That is, if the bearing configuration is similar to the one illustrated in Figure 3.1, radial bearings and thrust bearings are to be selected with their respective modules. Radial bearings are selected, first for the front end and then for the rear end. The flowcharts for these modules are given in Figures

3.6 and 3.7. If the spindle configuration is similar to the one illustrated in Figures 3.2 and 3.3, angular contact ball or taper roller bearings are selected and the module for this configuration is illustrated in Figure 3.8. The radial reactions at each end, axial force and the diameters of shafts at the front and rear ends, shaft speeds are the input to these modules. Required life, number of bearings at the front and rear ends, static strength and application factors are the additional input to these modules. After data is input, the program gives a message about suitable bearings for the given application. The user at this stage decides on the type and specifies the amount of preload for thrust ball, angular contact ball and taper roller bearings, or the user specifies the amount of interference or clearance for radial bearings such as cylindrical roller bearings or needle roller bearings.

Unless otherwise stated, the program will automatically select the bearings from among the available bearings database which includes all the data in manufacturers' catalogues.

All the bearings whose bore diameters are equal to the specified diameter are tested and the one which satisfies life requirement is selected. If all the bearings at the specified bore diameter do not satisfy life requirement, then the program advises the user either to try another bearing with one size larger or increase the number of bearings.

At the selection stage, the user may specify the designation of the bearing that she/he would consider to use and the computer will find all the parameters of the designated bearing from the bearing data base and test it for life requirement.

For the selected bearing(s), the stiffness' and deflections are calculated. If necessary, a new bearing may be selected or the program action is terminated.

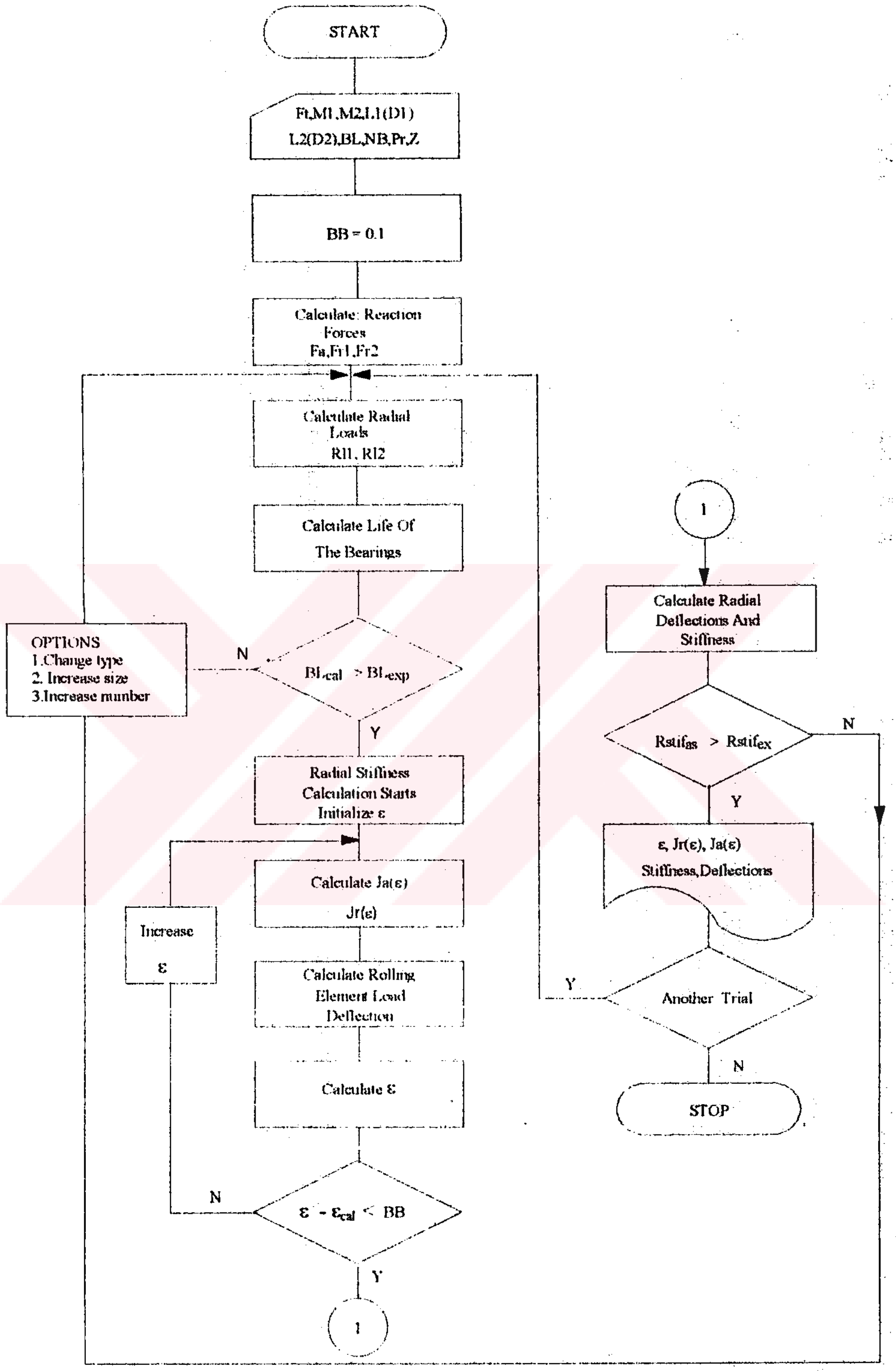


Figure 3.6 A Simplified Flowchart for Radial Bearing Selection [21]

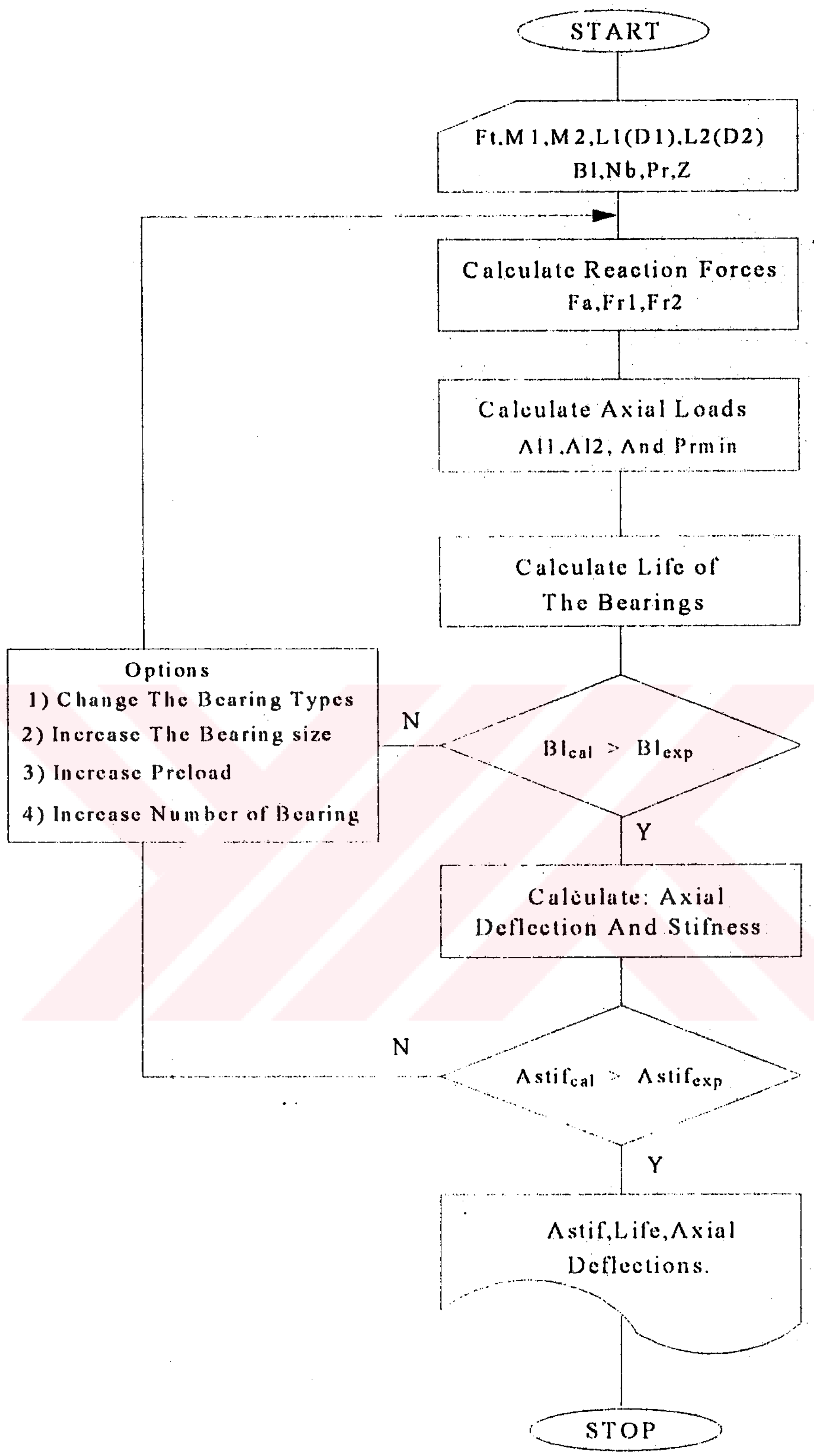


Figure 3.7 A Simplified Flowchart for Thrust Bearing Selection [21]

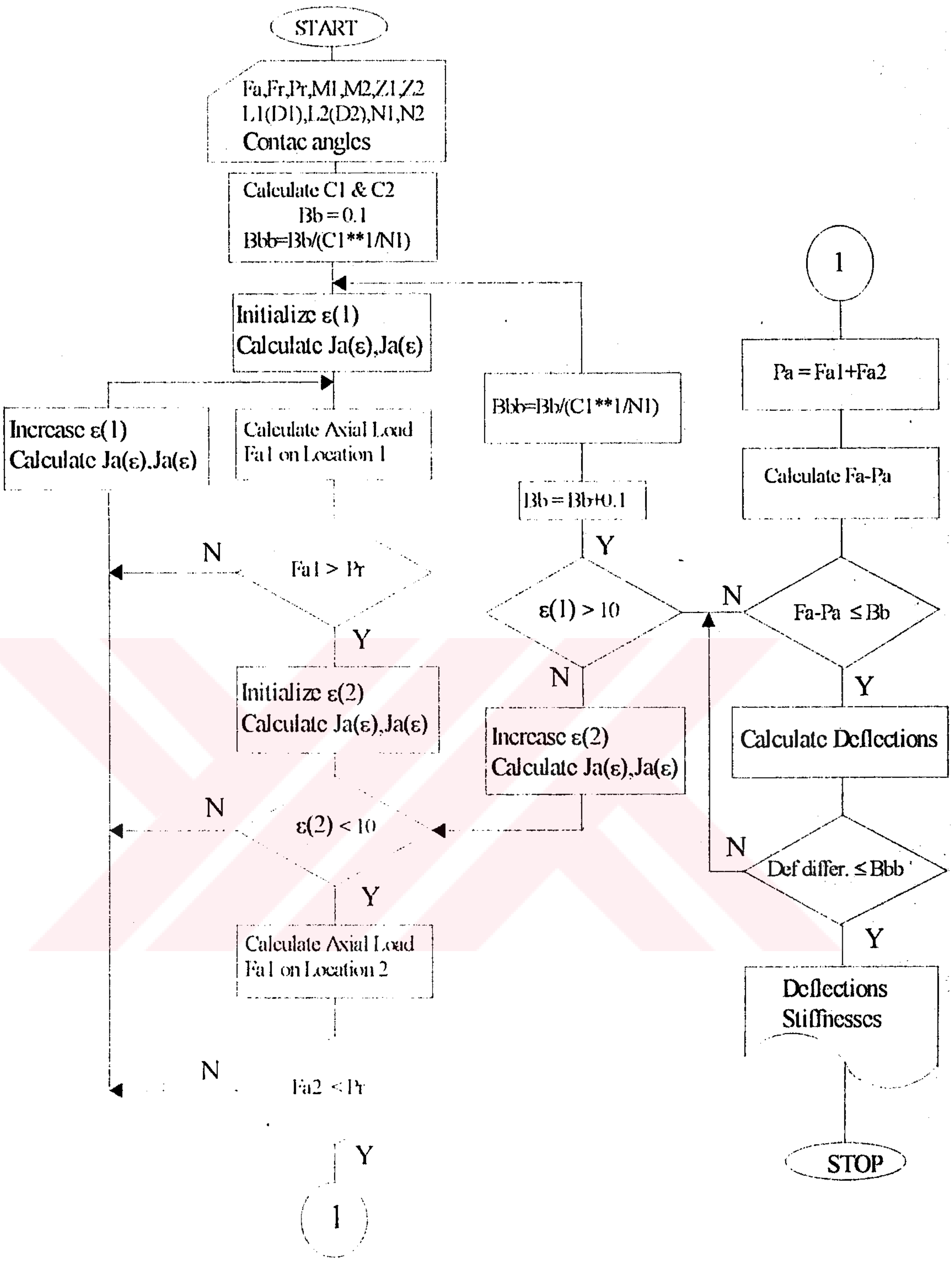


Figure 3.8 A Simplified Flowchart for Combined Radial/Axial Bearing Selection [21].

3.6.2 Examples

The application of analysis to bearing assemblies is verified through the examples by using the bearing sizing and selecting computer program. Two basic bearing configurations are considered. The program is arranged in such a way that a user may select a suitable bearing for a single loading condition or for the configurations described in the previous sections.

As the first example, the flow is given for a bearing selection for a single specified loading condition.

For bearing configuration I as illustrated in Figure 3.1, cylindrical roller bearings for both ends and thrust ball bearings for the front end are selected. The input data and flow of the program for the cylindrical roller bearing selection and for thrust ball bearing selection is illustrated below.

For bearing configuration II, either angular contact ball or taper roller bearings are selected. The flow of the program for the selection angular contact ball bearings is illustrated below.

Example 1: Consider a rolling element bearing is subjected to radial load of 3 kN and axial load of 5 kN. Select a suitable bearing having bore diameter of 60 mm based on a life of L_h=12000 hours at a speed of 1500 rpm. The bearing may sometimes be subjected to light shocks.

ANTIFRICTION BEARING SELECTION

Selection : Single bearing or Bearings on a shaft : S
Bore Diameter of Bearing (mm) : 60
Axial Load on Bearing (N) : 5000
Radial Load on Bearing (N) : 3000

Application Factors

Application	Factor
Belt drives	
V-Belts	2.0-2.5
Plain belts with tension pulleys	1.0-2.5
Plain belts	4.0-5.0
Type of machines	
Machines with shock free operation	1.0-1.2
Machines with minor shock loads	1.3-1.8
Machines with high shock loads	1.4-2.5
Machines with very high shock loads	1.8-3.0

Application Factor : 1.2

Static Strength Factors

Operating Conditions	Factor
Vibration free running	0.5
Average working conditions with normal demands on quiet running	1.0
Pronounced shock loads	1.5-2.0
High demands on quiet running	2.0
Spherical roller bearings >=	2.0

Static Strength Factor : 1
Expected life (hr) : 12000
Bearing Speed (rpm) : 1500
Type of Lubrication is Grease or Oil : O
Would you like to test a Special or All suitable types of Bearing : S

Suitable bearings for this application

1 -> Single Row Deep Groove Ball Bearing
2 -> Self Aligning Ball Bearing
3 -> Single Row Angular Contact Ball Bearing
4 -> Single Row Angular Contact Ball Bearing, in Pairs, Tandem
5 -> Single Row Angular Contact Ball Bearing, in Pairs, B-B, F-F
6 -> Double Row Angular Contact Ball Bearing, Series 32-33
7 -> Double Row Angular Contact Ball Bearing, Series 33D
8 -> Four Point Contact Ball Bearing, Duplex
13 -> Double Row Spherical Roller Bearing
14 -> Taper Roller Bearing

Bearing Number : 3

This bearing is not recommended for single use
Do you want to Continue or Not : N

Bearing Number : 6

Single Bearing Selection

Double Row Angular Contact Ball Bearing, Series 32-33
Bearing Designation : 3312
Bearing Diameter (mm) : 60
Calculated Bearing Life (h) : 14029.77
Static Force on Bearing (N) : 6150
Dynamic Force on Bearing (N) : 9252
Static Load Capacity (N) : 95000
Dynamic Load Capacity (N) : 100000
Calculated Static Strength (N) : 6150

Options Available

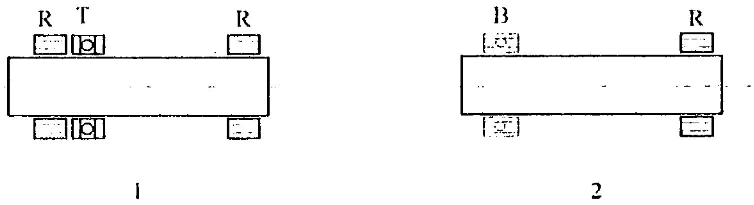
1-> To Input New Application
2-> To Check Input data
3-> To Change Bore Diameter
5-> To Change Axial Load
6-> To Change Radial Load
8-> To Change Application Factor
9-> To Change Static Strength Factor
10-> To Change Expected Life
11-> To Change Bearing Speed
12-> To Change Type of Lubrication
15-> To Exit

Enter Selection Number : 15

Example 2: Consider that a shaft is supported by cylindrical roller bearings at each end to carry radial loads and by thrust ball bearing at one end to carry axial load only. Radial reactions at left end and right end are 1250 and 6250 N, respectively and axial load is 2000 N. Select the suitable bearings having bore diameter of 35 mm based on a life of L_h=10000 hours at a speed of 1000 rpm.

ANTIFRICTION BEARING SELECTION

Selection : Single bearing or Bearings on a shaft : B
Axial load is carried by One end or Both end : O



R -> Radial Bearing, carries radial load only
T -> Thrust Bearing, carries thrust load only
B -> Bearing carries thrust and radial load.
Which configuration do you prefer : 1 or 2 : 1

Thrust bearing Selection

Bore Diameter of Bearing (mm) : 35
Number of Bearings : 1
Axial Load on Bearing (N) : 2000

Application Factors

Application	Factor
Belt drives	
V-Belts	2.0-2.5
Plain belts with tension pulleys	1.0-2.5
Plain belts	4.0-5.0
Type of machines	
Machines with shock free operation	1.0-1.2
Machines with minor shock loads	1.3-1.8
Machines with high shock loads	1.4-2.5
Machines with very high shock loads	1.8-3.0

Application Factor : 1.0

Static Strength Factors

Operating Conditions		Factor
Vibration free running		0.5
Average working conditions with normal demands on quiet running		1.0
Pronounced shock loads		1.5-2.0
High demands on quiet running		2.0
Spherical roller bearings	>=	2.0

Static Strength Factor : 1.0
Expected life (hr) : 10000

Bearing Speed (rpm) : 1000
Type of Lubrication is Grease or Oil : O
Would you like to test a Special or All suitable types of Bearing : S

Suitable bearings for this application

9 -> Thrust Ball Bearing, Single Direction
10 -> Thrust Ball Bearing, Double Direction
15 -> Cylindrical Thrust Roller Bearing
16 -> Needle Roller Thrust Bearing

Bearing Number : 10

Thrust Ball Bearing, Double Direction
Bearing Designation : 52209
Bearing Diameter (mm) : 35
Calculated Bearing Life (h) : 163223.9
Static Force on Bearing (N) : 2000
Dynamic Force on Bearing (N) : 2000
Static Load Capacity (N) : 68000
Dynamic Load Capacity (N) : 31500
Calculated Static Strength (N) : 2000

Options Available

2-> To Check Input data
3-> To Change Bore Diameter
5-> To Change Axial Load
8-> To Change Application Factor
9-> To Change Static Strength Factor
10-> To Change Expected Life
11-> To Change Bearing Speed
12-> To Change Type of Lubrication
15-> To Continue

Enter Selection Number : 15

Radial bearing Selection for Left end

Bore Diameter of Bearing (mm) : 35
Number of Bearings : 1
Radial Load on Bearing (N) : 1250
Would you like to test a Special or All suitable types of Bearing : S

Suitable bearings for this application

11 -> Cylindrical Roller Bearing
12 -> Needle Roller Bearing, with inner ring

Bearing Number : 11

Cylindrical Roller Bearing
Bearing Designation : 1007
Bearing Diameter (mm) : 35
Calculated Bearing Life (h) : 144986.4
Static Force on Bearing (N) : 1250
Dynamic Force on Bearing (N) : 1250
Static Load Capacity (N) : 11600
Dynamic Load Capacity (N) : 19000
Calculated Static Strength (N) : 1250

Options Available

2-> To Check Input data
3-> To Change Bore Diameter
5-> To Change Axial Load
6-> To Change Radial Load
8-> To Change Application Factor
9-> To Change Static Strength Factor
10-> To Change Expected Life
11-> To Change Bearing Speed
12-> To Change Type of Lubrication
15-> To Continue

Enter Selection Number : 15

Radial bearing Selection for Right end

Bore Diameter of Bearing (mm) : 35
Number of Bearings : 1
Radial Load on Bearing (N) : 6250
Would you like to test a Special or All suitable types of Bearing : A

Cylindrical Roller Bearing
Bearing Designation : 2207
Bearing Diameter (mm) : 35
Calculated Bearing Life (h) : 10323.1
Static Force on Bearing (N) : 6250
Dynamic Force on Bearing (N) : 6250
Static Load Capacity (N) : 29000
Dynamic Load Capacity (N) : 43000
Calculated Static Strength (N) : 6250

Needle Roller Bearing, with inner ring
Bearing Designation : 6907
Bearing Diameter (mm) : 35
Calculated Bearing Life (h) : 5709.398 Bearing Life is Inadequate.
Increase Bore diameter or change bearing type

Options Available

1-> To Input New Application
2-> To Check Input data

3-> To Change Bore Diameter
5-> To Change Axial Load
6-> To Change Radial Load
8-> To Change Application Factor
9-> To Change Static Strength Factor
10-> To Change Expected Life
11-> To Change Bearing Speed
12-> To Change Type of Lubrication
15-> To Exit

Enter Selection Number : 15

Example 3: Consider a particular shaft assembly which consists of two identical angular contact ball bearings in back-to-back arrangement subjected to 6 kN preload. Radial reactions at left end and right end are 3000 and 1280 N, respectively and axial load is 2000 N. Select the suitable bearings having bore diameter of 30 mm based on a life of L_h=10000 hours at a speed of 1000 rpm.

ANTIFRICTION BEARING SELECTION

Selection : Single bearing or Bearings on a shaft : B
Axial load is carried by One end or Both end : B

Back-to-Back Arrangement

Face-to-Face Arrangement

Arrangement is Back-to-Back or Face-to-Face : B
Bearing is Angular contact ball or Taper roller : A
The Inner or Outer ring is rotating : I
Direction of Axial load, Left to right or Right to left : L

Bearing Selection for Left end

Bore Diameter of Bearing (mm) : 30
Number of Bearings : 2
Number of Ball : 11
Ball Diameter (mm) : 28.6
Contact Angle (Deg) : 40
Axial Load on Bearing (N) : 2000
Radial Load on Bearing (N) : 3000

Application Factors

Application	Factor
Belt drives	
V-Belts	2.0-2.5
Plain belts with tension pulleys	1.0-2.5
Plain belts	4.0-5.0
Type of machines	
Machines with shock free operation	1.0-1.2
Machines with minor shock loads	1.3-1.8
Machines with high shock loads	1.4-2.5
Machines with very high shock loads	1.8-3.0

Application Factor : 1.0

Static Strength Factors

Operating Conditions	Factor
Vibration free running	0.5
Average working conditions with normal demands on quiet running	1.0
Pronounced shock loads	1.5-2.0
High demands on quiet running	2.0
Spherical roller bearings >=	2.0

Static Strength Factor : 1
Expected life (hr) : 10000
Bearing Speed (rpm) : 2000
Type of Lubrication is Grease or Oil : O

Bearing Selection for Right end

Bore Diameter of Bearing (mm) : 30
Number of Bearings : 1
Bearings are Identical or Not : I
Radial Load on Bearing (N) : 1280
The pair is Preloaded or Not : P
Assigned preload (N) : 0
It must be greater than 1308.11 . Press any key to continue
Assigned preload (N) : 6000

Bearing at Left End		Bearing at Right End	
D1 (mm) :	30	D2 (mm) :	30
N1 :	2	N2 :	1
Z1 :	11	Z2 :	11
C.Angle1 (Deg) :	40	C.Angle2 (Deg) :	40
Fa (daN) :	200	Pl (daN) :	600
Fr1 (daN) :	150	Fr2 (daN) :	128
Fa1 (daN) :	372.4728	Fa2 (daN) :	544.8222
Qmax1 (daN) :	90.37894	Qmax2 (daN) :	108.4922
Defmax1 (mm) :	1.317102E-02	Defmax2 (mm) :	1.487671E-02
Eps.1 :	1.555	Eps.2 :	2.365
Ja(Eps1) :	.582864	Ja(Eps2) :	.7102255

Jr(Eps2) : .1969597		Jr(Eps2) : .1400119	
AxiDef1 (mm) : 1.390189E-02		AxiDef2 (mm) : 1.825102E-02	
PrlDef1 (mm) : 1.237697E-02		PrlDef2 (mm) : 1.964722E-02	
RaDef1 (mm) : 5.528471E-03		RaDef2 (mm) : 4.105743E-03	
RaStf1 (N/mm) : 54264.55		RaStf2 (N/mm) : 31175.85	
AxStf1 (N/mm) : 1311545		AxStf2 (N/mm) : 1432454	

Bearing Selection for Left end

Single Row Angular Contact Ball Bearing
Bearing Designation : 7306
Bearing Diameter (mm) : 30
Calculated Bearing Life (h) : 6203.713 Bearing Life is Inadequate.
Increase Bore diameter or change bearing type

Bearing Selection for Right end

Single Row Angular Contact Ball Bearing
Bearing Designation : 7306
Bearing Diameter (mm) : 30
Calculated Bearing Life (h) : 3.829881 Bearing Life is Inadequate.
Increase Bore diameter or change bearing type

Options Available

1-> To Input New Application
2-> To Check Input data
3-> To Change Bore Diameter at Left end
4-> To Change Bore Diameter at Right end
5-> To Change Axial Load
6-> To Change Radial Load at Left end
7-> To Change Radial Load at Right end
8-> To Change Application Factor
9-> To Change Static Strength Factor
10-> To Change Expected Life
11-> To Change Bearing Speed
12-> To Change Type of Lubrication
13-> To Change Preload
14-> To Change Configuration
15-> To Exit

Enter Selection Number : 3
Bore Diameter of Bearing at Left end (mm) : 45

Bearing at Left End			Bearing at Right End		
D1	(mm) :	45	D2	(mm) :	30
N1	:	2	N2	:	1
Z1	:	11	Z2	:	11
C.Angle1	(Deg) :	40	C.Angle2	(Deg) :	40
Fa	(daN) :	200	Pl	(daN) :	600
Fr1	(daN) :	150	Fr2	(daN) :	128
Fa1	(daN) :	372.4728	Fa2	(daN) :	544.8222

Qmax1 (daN) :	90.37894	Qmax2 (daN) :	108.4922
Defmax1 (mm) :	1.317102E-02	Defmax2 (mm) :	1.487671E-02
Eps.1 :	1.555	Eps.2 :	2.365
Ja(Eps1) :	.582864	Ja(Eps2) :	.7102255
Jr(Eps2) :	.1969597	Jr(Eps2) :	.1400119
AxiDef1 (mm) :	1.390189E-02	AxiDef2 (mm) :	1.825102E-02
PrlDef1 (mm) :	1.237697E-02	PrlDef2 (mm) :	1.964722E-02
RaDef1 (mm) :	5.528471E-03	RaDef2 (mm) :	4.105743E-03
RaStf1 (N/mm) :	54264.55	RaStf2 (N/mm) :	31175.85
AxStf1 (N/mm) :	1311545	AxStf2 (N/mm) :	1432454

Bearing Selection for Left end

Single Row Angular Contact Ball Bearing
Bearing Designation : 7309
Bearing Diameter (mm) : 45
Calculated Bearing Life (h) : 40893.61
Static Force on Bearing (N) : 1718.429
Dynamic Force on Bearing (N) : 2648.095
Static Load Capacity (N) : 33500
Dynamic Load Capacity (N) : 45000
Calculated Static Strength (N) : 1718.429

Bearing Selection for Right end

Single Row Angular Contact Ball Bearing
Bearing Designation : 7306
Bearing Diameter (mm) : 30
Calculated Bearing Life (h) : 3.829881 Bearing Life is Inadequate.
Increase Bore diameter or change bearing type

Option Available

1-> To Input New Application
2-> To Check Input data
3-> To Change Bore Diameter at Left end
4-> To Change Bore Diameter at Right end
5-> To Change Axial Load
6-> To Change Radial Load at Left end
7-> To Change Radial Load at Right end
8-> To Change Application Factor
9-> To Change Static Strength Factor
10-> To Change Expected Life
11-> To Change Bearing Speed
12-> To Change Type of Lubrication
13-> To Change Preload
14-> To Change Configuration
15-> To Exit

Enter Selection Number : 15

CHAPTER IV

COMPUTER AIDED DESIGN OF JOURNAL BEARINGS

4.1. INTRODUCTION

In this chapter, computer aided design of journal bearings is studied. In section 4.2, Journal bearings are discussed in general. Hydrodynamically lubricated bearings are the types considered in this thesis.

The design method of the journal bearing is briefed in section 4.3. Section 4.4 is devoted to Computer aided design of journal bearings and the use of the computer program is illustrated with examples in this section.

4.2 JOURNAL BEARINGS

A journal bearing is a supporting member consisting of a rotating journal that is surrounded by a sleeve having a diameter slightly greater than that of the journal. There is an oil between the journal and the bearing. The bearing held fixed to prevent the radial motion of the shaft while shaft is rotating. A typical journal bearing is shown in Figure 4.1.

The dimension c is the radial clearance and is the difference in the radii of the bearing and journal. The centre of the journal is at 0 and the centre of the bearing at 0'. The distance between these centres is the eccentricity and is denoted by e. The minimum film thickness is designated by h_0 and it occurs at the line of centres. The film thickness at any other point is designated by h. We also define an eccentricity ratio ε as e/c. The quantity h_0/c is called the minimum film-thickness variable.

Depending upon the value of the angle β, these bearings may be classified as full bearing, partial bearing and fitted bearing.

The object of lubrication is to reduce friction, wear, and heating of machine parts which move relative to each other. A lubricant is any substance which, when inserted between the moving surfaces, accomplishes these purposes.

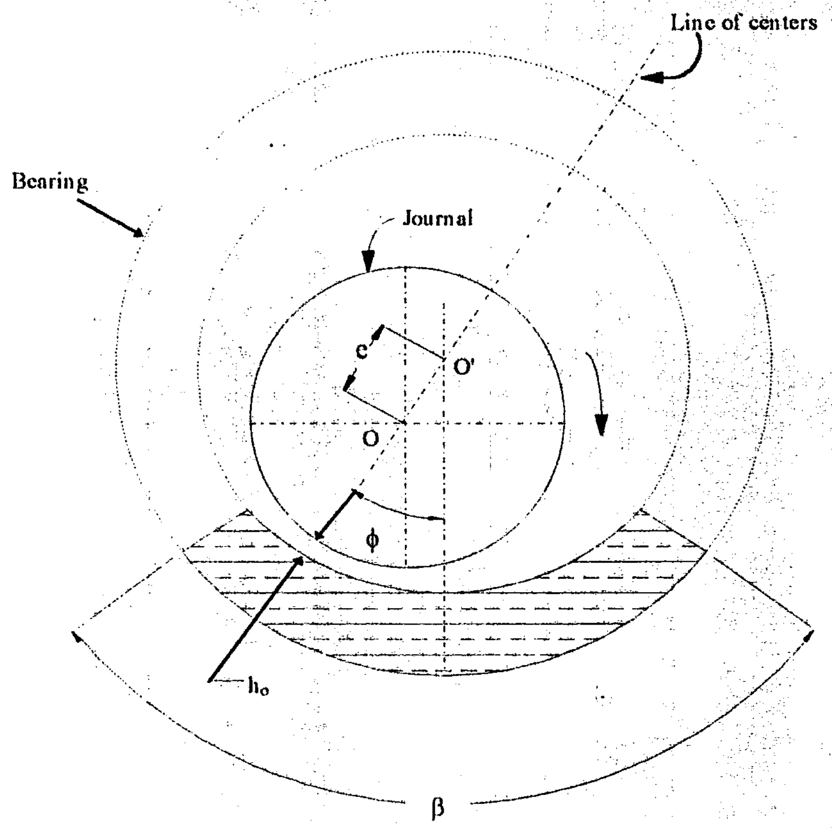


Figure 4.1. Nomenclature of a Journal Bearing

In a sleeve bearing, a shaft, or journal, rotates or oscillates within a sleeve, or bearing, and the relative motion is sliding. In an anti-friction bearing, the main relative motion is rolling.

The field of application for journal bearing is immense. The crankshaft and connecting-rod bearings are at high temperatures and under varying load conditions. The journal bearing used in the steam turbines of power-generating stations are said to have reliability's approaching 100 percent. At the other extreme there are thousands of applications in which the loads are light and the service relatively unimportant.

4.3 DESIGN OF JOURNAL BEARINGS

In the classical design methods selection of the parameters are mainly based on several assumptions and trial-error procedures. The design procedures are therefore, iterative and continues until the initial assumptions are satisfied.

The variables which are involved in the design of journal bearing may be grouped under two headings as design variables and performance variables.

a) Design variables:

The rotational speed N,

The load per unit of projected bearing P,

The type of oil

Bearing dimensions

This group of variables are either specified by the overall design of the machine or under the control of designer.

b) Performance variables:

The coefficient of friction f

The temperature rise ΔT

The flow rate of lubricant Q

The maximum film pressure P_{max}

The minimum film thickness h_o.

The important problem in bearing design is to define satisfactory limits for dependent variables (performance variables) and then to decide on the first group of variables so that these limitations are not exceeded.

The relationships between all these parameters are difficult to obtain analytically. Raimondi and Boyd [30] used iteration technique to solve Reynolds's equation and made available the necessary data for the design of journal bearings. They create charts for different variables and we shall employ these charts which can be found in any textbook on mechanical engineering design. In these charts some relational variables are plotted against Sommerfeld number which is also known as bearing characteristic number for length-diameter ratios (l/d) of 1/4, 1/2, 1. β angles may vary between 60 and 360, but in this thesis we stick to the full (β=360°) bearing.

Sommerfeld number is defined by the equation;

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P} \tag{4.1}$$

where r: journal radius

c: radial clearance

μ: absolute viscosity

N: rotational speed (rps)

P: load per unit of projected bearing area

Viscosity, in the above expression, is the viscosity which corresponds to the average of the temperature of the oil at the inlet and the outlet. This creates problem in the design of journal bearings, since average temperature can not initially be known. Average temperature can be determined by using trial and error method, therefore the design process must be computerised for quick solutions. In the design study of journal bearings, one must always keep in mind that under severe loading conditions, there is a possibility of having metal to metal contact. Hydrodynamic lubrication is important from performance and wear points of view. Hydrodynamic lubrication means that the load carrying surface of the bearing are separated by 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.

Insufficient surface area, a drop in the velocity of the moving surface, a lessening in the quantity of lubricant delivered to a bearing, an increase in the bearing load or an increase in lubricant temperature resulting in a decrease in viscosity-any one of these -may prevent the build-up of a film thick enough for full-film lubrication. When this happens, the highest asperities may be separated by lubricant films only several molecular dimensions in thickness. This is called boundary lubrication. Friction level is very important in these bearings. This is why boundary lubrication must be avoided.

The torque required to overcome friction is

$$T = fWr \tag{4.2}$$

The power loss in the bearing:

$$H = 2\pi NT \tag{4.2}$$

Temperature rise of the oil is calculated by Shigley [1] and given in the following;

$$\Delta T = \frac{8.3P(r/c)f}{\left[1 - \frac{1}{2}\frac{Q_s}{Q}\right](Q/rcNl)} \tag{4.4}$$

In order to start with the design of journal bearings, the load to be carried, the rotational speed of the shaft, either the length or the diameter of the bearing must be specified. Recommended values of unit load given in Mechanical Engineering Handbooks, gives the designer an idea about the size of the bearing, the material and the manufacturing method of the journal and the bearing gives the designer an idea about the clearance or clearance ratio r/c. The designer is free to choose type of oil to be used.

For the determination of ΔT, frictional variable (r/c)f, flow variable Q/rcNl, and side flow Q_s/Q must be known. These variables are the functions of l/d ratio and Sommerfeld number which in turn is the function of viscosity corresponding to average temperature. Therefore the process is an iterative process. Initially, the temperature increase is assumed and then viscosity is determined to calculate Sommerfeld number. After that, variables are found from relevant charts and ΔT is calculated. Average temperature is then calculated from $T_{av} = T_i + \Delta T/2$.

The process will continue until calculated ΔT would be equal to the assumed ΔT. After the evaluation of performance factors, the designer will decide whether to stop the procedure or try with another set of design parameters.

4.4 COMPUTER AIDED DESIGN OF JOURNAL BEARINGS

4.4.1 Computer Program

The calculations involved in the design of journal bearings are of iterative type. The design algorithm mentioned above is automated with a computer program whose simplified flowchart is given in Figure 4.2. As can be seen from the flowchart, two options are made available to the designer. The first one is used to solve specific problems which the design variables under control of designer are known and performance variables are required. The second option deals with optimization of design, which is the main purpose, of this study also consists of two parts. The first one is optimum design based on bearing clearance which the bearing length is taken as constant and clearance is variable, as the clearance varies, performance variables are calculated. The second one is the optimum design based on bearing length which clearance is taken as constant and length is variable. As the length is varied, the performance variables are calculated. The selection of optimum region for both cases is left to the designer. The designer could not get both table and graph of the calculated performance variables as an output of the program to help the designer for decision of optimum region. The designer can compare several optimal solution without exiting from program.

Full journal bearings, operating at constant speed are investigated in the present study. The computer program helps us to investigate different options in a very short time. It helps the designer or user to see different design parameters and the time spend for selection of parameters decreases in a considerable amount.

The program can also make calculation for the desired value of l/d ratio with in the interval $\infty > L/d > 1/4$ according to interpolation equations given by Raimondi and Boyd [30].

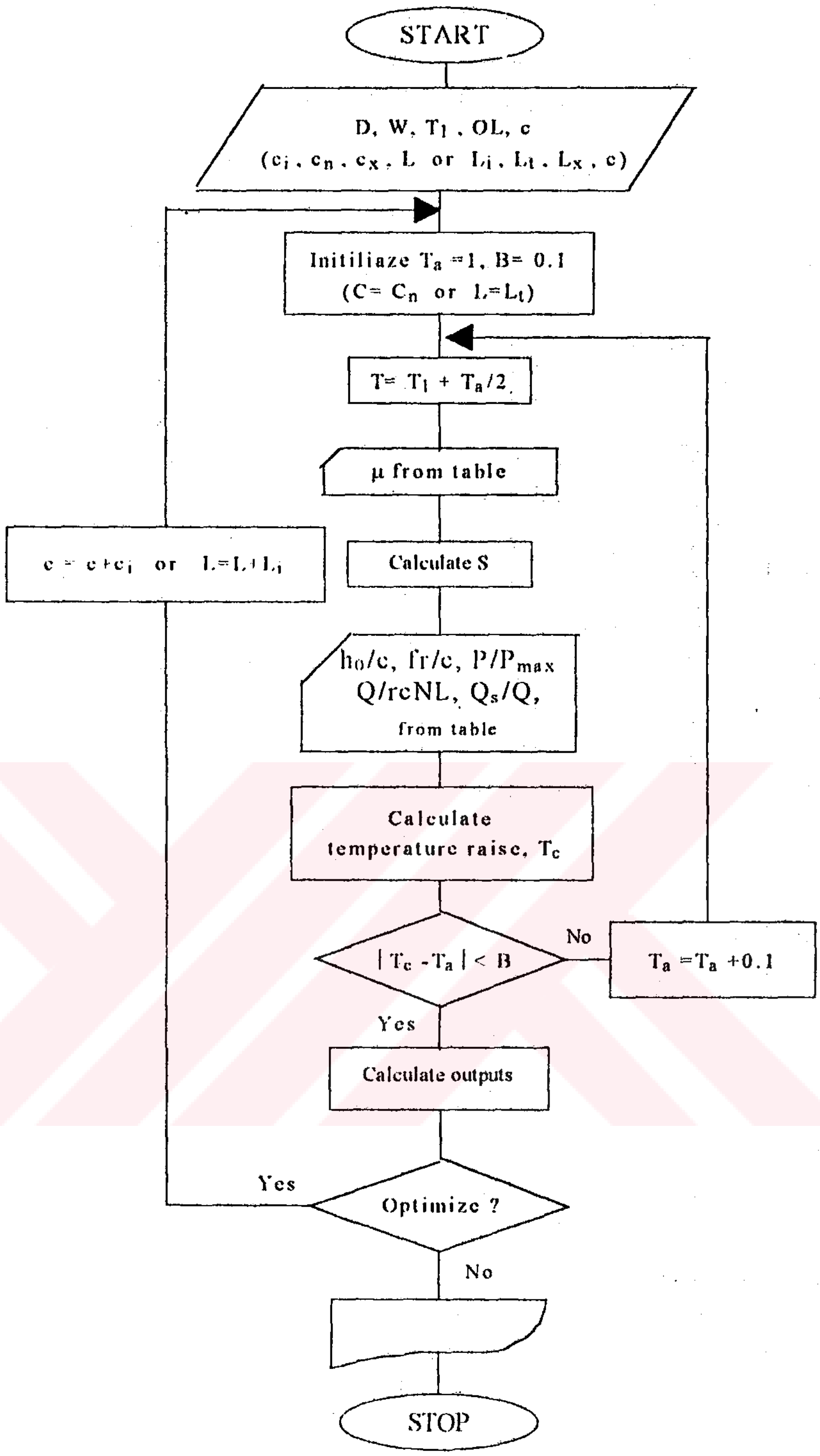


Figure 4.2 A Simplified Flowchart for the Design of Journal Bearings

Maximum load carrying capacity or minimum friction are considered as the criteria that may be used in the design study. In one of the charts created by Raimondi and Boyd [30], curves for these criteria are given and it is up to designer to use one of these curves.

When the program is executed, the input data is given and the program directs the user to reach his goal for the specified problem.

4.4.2 Examples

The use of the program is verified through the examples for different applications. As will be appreciated, it is not possible to show all the capabilities of the program in this thesis due to space limitation. Two examples are prepared. Example 1 is given for the illustration of the flow of the program for a specific case. Example 2 is given to show the variations of performance factors with respect to geometrical parameters.

Example 1: A journal bearing has a diameter of 50 mm and it is to operate at a speed of 1200 rpm and carry a load of 5 kN. The unit load is 2 MPa and the radial clearance is 48 μm. If SAE 30 oil at an inlet temperature of 50 C is used, determine the total flow and the amount of power loss in the bearing.

Bearing Diameter (mm) : 50
Radial Load on Bearing (N) : 5000
Journal Speed (rpm) : 1200
Type of Oil SAE : 30
Inlet Temperature (C) : 50
This is a Specific run or Not : S

Application	Unit Load
Diesel engines :	(MPa)
Main bearings	6 - 12
Crankpin	8 - 15
Wirstpin	14 - 15
Electric motors	0.8 - 1.5
Steam turbines	0.8 - 1.5
Gear reducers	0.8 - 1.5
Automotive engines:	
Main bearings	4 - 5
Crankpin	10 - 15
Air compressors:	
Main bearings	1 - 2
Crankpin	2 - 4
Centrifugal pumps	0.6 - 1.2

Recommended unit load (MPa) : 2

Up to Shaft diameter mm	-15	15 - 25	25 - 50	50 - 90	90 - 140
A-> 0.2-0.4 μm-rms	.006-.019	.019-.038	.038-.06	.06-.09	.09-.127
B-> 0.4-0.8 μm-rms	.013-.025	.025-.051	.051-.08	.08-.11	.11-.165
C-> 0.4-0.8 μm-rms	.013-.038	.025-.051	.038-.09	.05-.10	.08-.152
D-> 0.8-1.6 μm-rms	.051-.101	.064-.114	.076-.13	.10-.18	.13-.203
E-> 1.6-3.9 μm-rms	.076-.152	.127-.228	.203-.31	.28-.41	.36-.508
Automotive crankshaft			.038-.06	.06-.09	

A->Precision spindle,hardened ground,lapped into bronze bushing; V<180 mps
B->Precision spindle,hardened ground,lapped into bronze bushing; V>180 mps
C->Electric motors,generators,ground journal in broached or reamed bronze
D->General Machinery,turned or cold rolled journal in bored-reamed bronze
E->Rough Machinery,turned or cold rolled journal in poured babbit bearing

Radial clearance (mm) : .048

Process	Roughness: Usual	Extreme(rms)
Shaping	15-1.5	25-0.4
Drilling	6-1.5	25-0.4
Milling	6-0.8	25-0.2
Broaching	3-0.8	6-0.4
Reaming	3-0.8	6-0.4
Lathe work	6-0.4	25-0.05

Grinding	1.5-0.1	6-0.02
Polishing	0.4-0.1	0.8-0.01
Lapping	0.4-0.05	0.8-0.01

Surface roughness of journal (rms) : 3
Surface roughness of bearing (rms) : 4
Calculated L/D : 1
Do you want to Change L/D ratio or Not : N

Input Data

Bearing diameter	(mm) :	50.00
Radial load on bearing	(N) :	5000.00
Journal speed	(rpm) :	1200.00
Type of oil	SAE :	30
Inlet temperature of oil	(C) :	50.0
Recommended unit load	(MPa) :	2.000
Radial clearance	(mm) :	0.0480
Surface roughness of journal	(rms) :	3.000
Surface roughness of bearing	(rms) :	4.000

Output Data

L/D ratio	:	1.00
Calculated Length	(mm) :	50.00
Calculated unit load	(MPa) :	2.000
Flow rate of oil	(mm^3/s) :	5347.9
Side flow rate of oil	(mm^3/s) :	4035.7
Minimum film thickness	(mm) :	0.015
Outlet temperature of oil	(C) :	64.4
Maximum pressure in bearing	(MPa) :	5.295
Power loss due to friction	(Hp) :	0.098

Press any key to continue

Option Available

1-> To Input New Values
2-> To Check Input Data
3-> To Change Bearing Diameter
4-> To Change Radial Load
5-> To Change Journal Speed
6-> To Change Type of Oil
7-> To Change Inlet Temperature
8-> To Change L/D Ratio
9-> To Change Recommended unit load
10-> To Change Radial clearance
11-> To Change Surface roughness of journal
12-> To Change Surface roughness of bearing
21-> To EXIT

Enter Selection Number : 6

Type of Oil SAE : 40

Input Data

Bearing diameter	(mm) :	50.00
Radial load on bearing	(N) :	5000.00
Journal speed	(rpm) :	1200.00
Type of oil	SAE :	40
Inlet temperature of oil	(C) :	50.0
Recommended unit load	(MPa) :	2.000
Radial clearance	(mm) :	0.0480
Surface roughness of journal	(rms) :	3.000
Surface roughness of bearing	(rms) :	4.000

Output Data

L/D ratio	:	1.00
Calculated Length	(mm) :	50.00
Calculated unit load	(MPa) :	2.000
Flow rate of oil	(mm^3/s) :	5228.2
Side flow rate of oil	(mm^3/s) :	3677.5
Minimum film thickness	(mm) :	0.018
Outlet temperature of oil	(C) :	67.3
Maximum pressure in bearing	(MPa) :	4.974
Power loss due to friction	(Hp) :	0.119

Press any key to continue

Option Available

1-> To Input New Values
2-> To Check Input Data
3-> To Change Bearing Diameter
4-> To Change Radial Load
5-> To Change Journal Speed
6-> To Change Type of Oil
7-> To Change Inlet Temperature
8-> To Change L/D Ratio
9-> To Change Recommended unit load
10-> To Change Radial clearance
11-> To Change Surface roughness of journal
12-> To Change Surface roughness of bearing
21-> To EXIT

Enter Selection Number : 21

Example 2: A journal bearing has a diameter of 35 mm and it is to operate at a speed of 1800 rpm and carry a load of 2225 N. If SAE 20 oil at an inlet temperature of 35 C is used and bearing length is taken as 35 mm, optimise the bearing clearance.

JOURNAL BEARING DESIGN

Bearing Diameter (mm) : 35
Radial Load on Bearing (N) : 2225
Journal Speed (rpm) : 1800
Type of Oil SAE : 20
Inlet Temperature (C) : 35
This is a Specific run or Not : N
The run is Based on bearing Length or Clearance : C

Up to Shaft diameter mm	-15	15 - 25	25 - 50	50 - 90	90 - 140
A-> 0.2-0.4 μm-rms	.006-.019	.019-.038	.038-.06	.06-.09	.09-.127
B-> 0.4-0.8 μm-rms	.013-.025	.025-.051	.051-.08	.08-.11	.11-.165
C-> 0.4-0.8 μm-rms	.013-.038	.025-.051	.038-.09	.05-.10	.08-.152
D-> 0.8-1.6 μm-rms	.051-.101	.064-.114	.076-.13	.10-.18	.13-.203
E-> 1.6-3.9 μm-rms	.076-.152	.127-.228	.203-.31	.28-.41	.36-.508
Automotive crankshaft			.038-.06	.06-.09	

A->Precision spindle,hardened ground,lapped into bronze bushing; V<180 mps
B->Precision spindle,hardened ground,lapped into bronze bushing; V>180 mps
C->Electric motors,generators,ground journal in broached or reamed bronze
D->General Machinery,turned or cold rolled journal in bored-reamed bronze
E->Rough Machinery,turned or cold rolled journal in poured babbit bearing

Initial value of Bearing clearance (mm) : .025
Final value of Bearing clearance (mm) : .085
Increment value of Bearing clearance(mm) : .002
Bearing length (mm) : 35

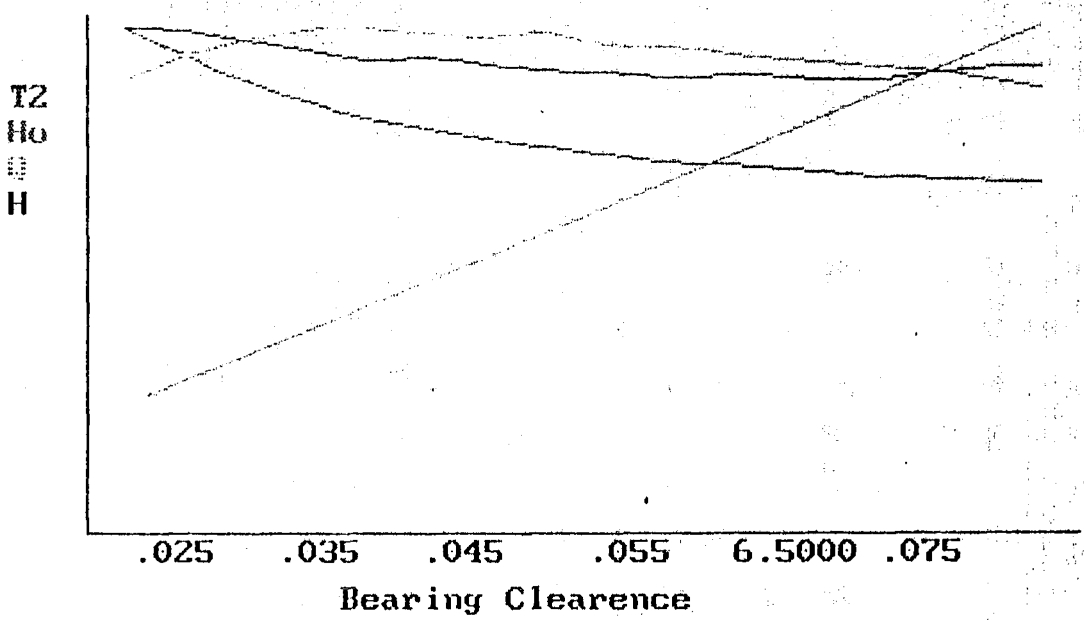
L (mm)	c (mm)	T_2 (C)	Q (mm^3/s)	Q_S (mm^3/s)	H_0 (mm)	H (J/s)	P (MPa)	P_{MAX} (MPa)
35.00	0.025	62.741	1842.864	948.063	0.014	57.492	1.816	3.887
35.00	0.027	60.860	2013.210	1094.518	0.015	57.397	1.816	3.904
35.00	0.029	59.198	2184.418	1244.173	0.015	57.238	1.816	3.948
35.00	0.031	57.518	2359.802	1395.502	0.015	56.663	1.816	4.042
35.00	0.033	56.038	2535.132	1547.443	0.016	56.103	1.816	4.130
35.00	0.035	54.696	2714.587	1707.396	0.016	55.493	1.816	4.204
35.00	0.037	53.530	2897.706	1873.717	0.016	55.012	1.816	4.262
35.00	0.039	52.512	3075.464	2045.540	0.016	54.423	1.816	4.340
35.00	0.041	51.600	3251.628	2218.447	0.016	53.845	1.816	4.417
35.00	0.043	50.915	3434.256	2397.921	0.016	53.861	1.816	4.492
35.00	0.045	50.323	3617.933	2578.553	0.016	54.023	1.816	4.562

35.00	0.047	49.676	3799.091	2758.155	0.016	53.773	1.816	4.634
35.00	0.049	48.978	3976.926	2934.883	0.016	53.109	1.816	4.705
35.00	0.051	48.391	4163.584	3119.247	0.016	52.795	1.816	4.775
35.00	0.053	47.877	4351.346	3303.671	0.016	52.630	1.816	4.839
35.00	0.055	47.396	4531.295	3485.039	0.016	52.338	1.816	4.933
35.00	0.057	46.979	4709.479	3662.276	0.016	52.201	1.816	5.021
35.00	0.059	46.584	4891.748	3860.185	0.015	51.944	1.816	5.088
35.00	0.061	46.246	5074.008	4059.818	0.015	51.829	1.816	5.141
35.00	0.063	45.922	5256.557	4259.944	0.015	51.702	1.816	5.196
35.00	0.065	45.664	5438.683	4442.263	0.015	51.951	1.816	5.292
35.00	0.067	45.368	5620.422	4623.566	0.015	51.939	1.816	5.384
35.00	0.069	45.029	5801.811	4803.931	0.015	51.625	1.816	5.471
35.00	0.071	44.758	5982.247	4989.561	0.015	51.522	1.816	5.550
35.00	0.073	44.507	6163.238	5183.512	0.015	51.407	1.816	5.630
35.00	0.075	44.311	6343.944	5376.406	0.015	51.536	1.816	5.691
35.00	0.077	44.185	6524.388	5568.316	0.015	52.010	1.816	5.726
35.00	0.079	44.058	6705.344	5762.564	0.015	52.443	1.816	5.762
35.00	0.081	43.887	6887.557	5949.360	0.015	52.647	1.816	5.872
35.00	0.083	43.707	7070.697	6137.868	0.014	52.753	1.816	5.997
35.00	0.085	43.517	7263.840	6332.517	0.014	52.837	1.816	6.091

Option Available

1-> To Input New Values
2-> To Check Input Data
3-> To Change Bearing Diameter
4-> To Change Radial Load
5-> To Change Journal Speed
6-> To Change Type of Oil
7-> To Change Inlet Temperature
17-> To Change Initial value of Bearing Clearence
18-> To Change Final value of Bearing Clearence
19-> To Change Increment value of Bearing Clearence
20-> To Change Bearing Length
21-> To EXIT

Enter Selection Number : 21



CHAPTER V

DESIGN OF BOLTED JOINTS

5.1. INTRODUCTION

This chapter is devoted to computer aided design of bolted joints. Bolted joints are discussed in section 5.2. The methods of evaluation of bolt stiffness and member stiffness are explained in section 5.3. The relationship between external load, preload, number of bolts and the strength of bolted joint under static and fatigue loading condition are also included in this section. Section 5.4 is devoted to computer aided design of bolted joints and the use of the computer program is illustrated with examples in this section.

5.2. CONSIDERATIONS ON BOLTED JOINTS

The joining of removable machine and structural element is commonly accomplished by means of threaded bolts and screws. In other word, when a connection is desired which can be disassembled without destructive methods and which is strong enough to resist both external tensile loads and shear loads, or a combination of these, than the simple bolted joint using hardened washers is a good solution. The selection of the proper size and the number of bolts depend on many considerations.

Strength of bolts is specified in terms of tensile test of the threaded section. The load-elongation characteristics of a bolt are more significant than the stress-strain diagram of the parent metal, because performance is controlled by threads. Also the stress varies along the bolt as a result of the gradual introduction of force from the nut and the change in section from the threaded to the unthreaded portion. The weakest section of any bolt in tension is the threaded portion. When a number of bolts are employed in fastening two parts of a machine, such as a cylinder and cylinder head, the load carried by each bolt depends on its relative tightness. The tighter bolts carry the greater loads. On the other hand, it may be desirable to have the bolts the weakest part of the machine, since their breakage from overload in the machine will result in a minimum cost of replacement. In such cases, the breaking load of the bolts will be equal to the load which causes the connected members to be stressed up to elastic limit.

The selection of proper size of the bolt depends on such considerations as security, durability, vibration, consistent proportions and finish. When bolts are used to make a tight joint between two parts, an initial tension is induced in the bolt due to tightening. If the bolt is more yielding than the connecting members, it should be designed to resist the initial tension or the external load which is greater. In cases where the bolts are subjected to cyclic loading, an increase in the initial tightening load decreases the operating stress range.

5.3 THEORY OF BOLTED JOINTS

A typical bolted joint is shown in Figure 5.1. The design of a bolted joint requires determination of the number and the size of bolts as well as dimensions of the members to satisfy both sealing and strength requirements. That is, there must not be joint separation and static or fatigue failure of bolts under working load. In other words, there must always exist compression load on the members and the maximum load on the bolt must not be greater than the safe load.

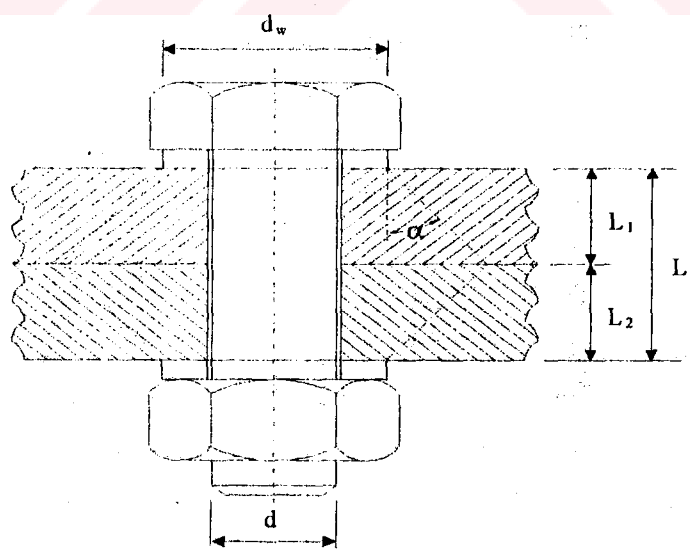


Figure 5.1 A Typical Bolted Joint

If a preload is applied to the joint a tensile load on the bolt and a compressive load on the members are induced. When an external load is applied on a preloaded connection, tensile load on the bolt will increase while some of the compression load on the members will be relieved. The manner of load sharing is dependent on the stiffness of the bolt and the member. The bolt and member stiffness, k_b and k_m, must be determined accurately to have an agreement between the theoretical analysis and the actual behaviour of joints. Because, overestimation of member stiffness will result smaller load on the bolt than the actual case which may result shorter fatigue life than expected while underestimation of the member stiffness will result smaller load on the members than the actual case which may result separation of the joint although not expected [31].

As agreed by all researchers, the stiffness of the bolt is calculated by considering it as a bar in simple tension.

$$k_b = \frac{AE_b}{L} = \frac{\pi E_b d^2}{4L} \tag{5.1}$$

Determination of the member stiffness, however, is complex because the compression spreads out between the bolt head and the nut and the compressed area is not uniform as shown in Figure 5.1 [31].

There have been many attempts to determine the member stiffness, k_m. As Fritsche reported [32], Rötscher is the first who proposed a conical stress distribution on the connected parts about the mid plane of the joint. He assumed a cone angle of 45^o but in his formulation he replaced conical frusta with a cylinder having the same cross sectional area. Shigley [1] pointed out that cone angle may be left as a variable during integration process but in their calculations, they have recommended to use $\alpha=30^{0}$ and find the member stiffness as;

$$k_m = \frac{0.577\pi E d}{\ln \dfrac{(1.15t + D - d)(D + d)}{(1.15t + D + d)(D - d)}} \tag{5.2}$$

where E is the modulus of elasticity of a member, d is the bolt diameter, D is the smaller diameter of conical frusta and t is the thickness of the member.

Yıldırım [33] and Filiz and Yıldırım [34], from their experimental work, suggested a stiffness formula which includes thickness ratio of the members. Filiz and Akpolat [35], had made considerable contribution for the evaluation of part stiffness by using Finite Element Method and suggested a formula which includes the effects of different materials and thickness', that is;

$$k_m = \frac{\pi}{2} d E_{eq} e^{\left(\frac{\pi}{5} - \beta_1\right) d/L} \frac{1}{1 - \beta_2} \tag{5.3}$$

where $\beta_1=(0.1d/L)^2$ is the correction factor for d/L and $\beta_2 = (1-L_1/L_2)^8$ is the correction factor for thickness ratio (L_1/L_2) and $E_{eq}= E_1E_2/(E_1+E_2)$. In this equation thickness ratio is less equal to unity and the member with smaller thickness is to be considered as the first member. This equation includes individual member thickness' and the ratio of bolt diameter to grip length besides the bolt diameter and modulus of elasticity of member materials [31].

If there are more than one members included in the grip of the bolt, the stiffness constant of members can be considered as a compressive springs in series, that is;

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \cdots + \frac{1}{k_i} \tag{5.4}$$

The effect of the preload is to place the bolted member components in compression for better resistance to external tensile load and to create a friction force between the parts to resist the shear load. When the external load P is applied to the preloaded assembly, the resultant load on the bolt F_b and the resultant force on the members F_m are determined from compatibility of deformations as:

$$F_b = CP + F_i \tag{5.5}$$

and

$$F_m = (1-C)P - F_i \tag{5.6}$$

where F_i is the preload on bolted joint due to tightening and the constant C is defined as;

$$C = \frac{k_b}{k_b + k_m} \tag{5.7}$$

If the external force is large enough, the members will separate and the entire load will be carried by the bolt.

The gasket may be used for preventing the members leakage in pressure vessel applications. The preload must satisfy the relation for the gasketed joint.

$$F_i \geq A_g P_o \tag{5.8}$$

where A_g gasket area, P_o minimum gasket seal pressure and in a gasketed joint the clamping load must satisfy the relation

$$F_m \geq m A_g P \tag{5.9}$$

where m is called the gasket factor, vary from about 2 to 4 for most material. It can be considered as a factor of safety. P is the gasket pressure.

Torque requirement can be calculated by using the formula;

$$T = 0.2 F_i d \tag{5.10}$$

For the plated nuts;

$$T = 0.15 F_i d \tag{5.11}$$

The proof load of a bolt F_p is the maximum load that a bolt can withstand without acquiring a permanent set and calculated from;

$$F_p = A_t S_p \tag{5.12}$$

where A_t is the tensile stress area and S_p is the proof strength which is the limiting value of the stress obtained from material table. The relationship between yield strength and proof strength is given as:

$$S_p = 0.85 S_y \tag{5.13}$$

It is recommended for both static and fatigue loading the following be used for preload.

$$F_i = 0.75 F_p \quad \text{for re-used connections}$$

$$F_i = 0.90 F_p \quad \text{for permanent connections} \tag{5.14}$$

For static loading case, the two important criteria are expressed as following paragraphs.

For there is no joint separation the load on the members must be greater than zero and for there is no strength problem. the stress on the bolt must be smaller than the proof strength of the bolt. If both criteria are combined, the following inequality could be written:

$$n(1-C)\,P < Fi < At\,Sp - n\,C\,P \tag{5.15}$$

where n is considered to be load factor of safety. Most of the time the type of fatigue loading encountered in the analysis of the bolted joints is one in which the externally applied load fluctuates between zero and some maximum force P. However, in order to determine the mean and alternating bolt stresses:

$$\sigma_a = \frac{CP}{2A_t} \quad \text{and} \quad \sigma_m = \frac{CP}{2A_t} + \frac{F_i}{A_t} \tag{5.16}$$

When we substitute this alternating and mean bolt stress into modified Goodman criterion of failure and solve for Fi, then,

$$F_i = A_t S_{ut} - \frac{nCP}{2N}\left(\frac{S_{ut}}{S_e} + 1\right) \tag{5.17}$$

where N is the number of bolts used, S_{ut} is the ultimate tensile strength and S_e is the endurance limit for machine element.

5.4. COMPUTER AIDED DESIGN OF BOLTED JOINTS

5.4.1 Computer Program

The calculations involved in the design of bolted joints are of iterative type. For different applications and different requirements, there are so many considerations that must be taken into account, a sound design could be difficult to obtain in the possible shortest time without computers.

A design algorithm is developed and it is automated with a computer program whose simplified flowchart is given in Figure 5.2.

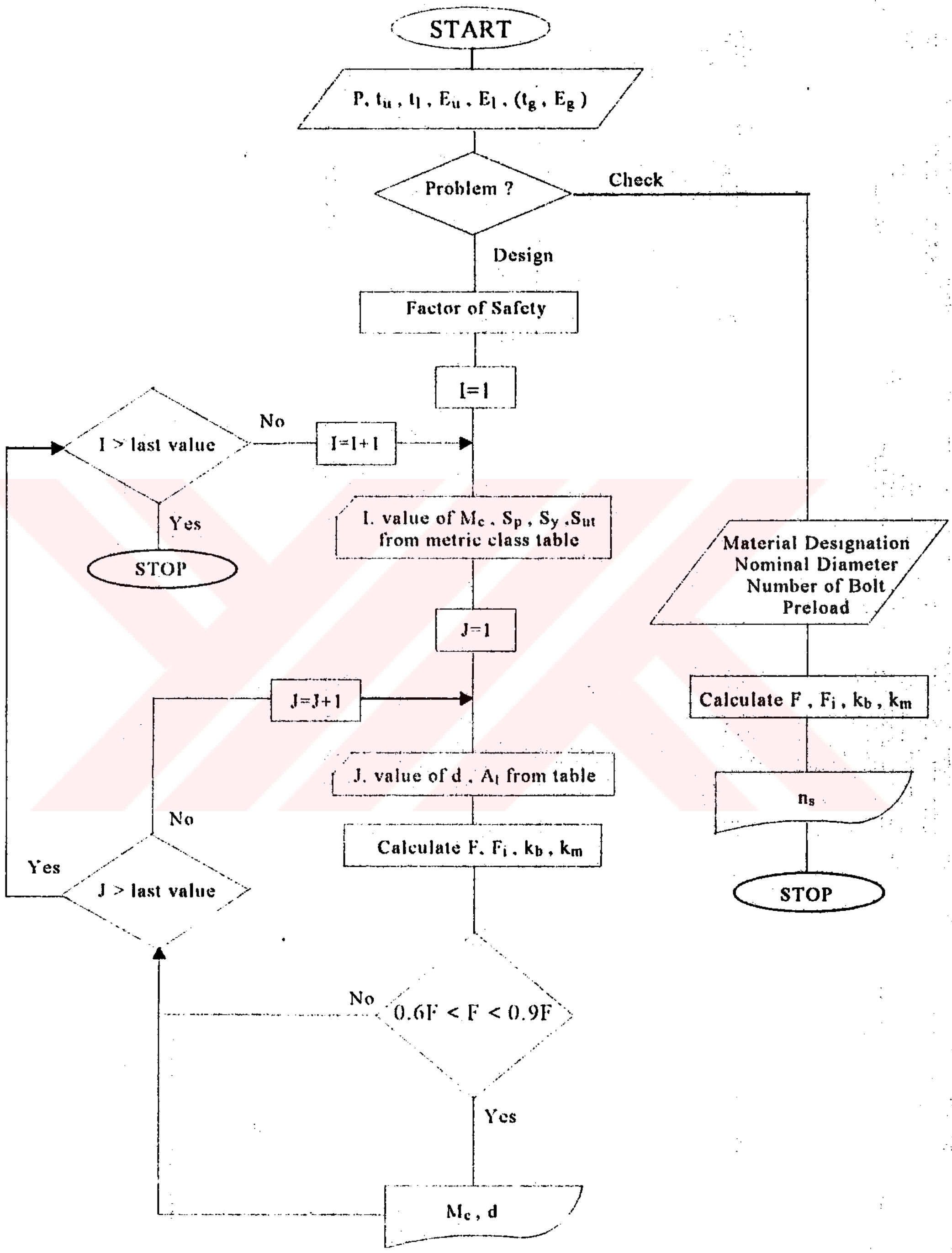


Figure 5.2 A Simplified Flowchart of the Program for Bolted Joint Design

As can be seen from the flowchart, the program could be used for the calculation of factor of safety if all parameters are specified at the beginning or to determine suitable number of bolts for the joint whose geometry and factor of safety are specified. A single bolted joint similar to given in Figure 5.1 or joints for pressure vessel or similar applications can be tested with this program.

When the program is executed, the input data is given and the program directs the user to reach his goal for the specified problem.

5.4.2 Examples

The use of the program is verified through the examples for different applications. As will be appreciated, it is not possible to show all the capabilities of the program in this thesis due to space limitation. Two examples are prepared. The flow of the program is given for both of these examples.

Example 1: A bolted connection having 4 bolt is to be used to resist a separating force of 10 kN. Members have same thickness of 16 mm and modulus of elasticity of 80 GPa. If bolt diameter is 8 mm, determine design factor.

BOLT DESIGN

```
Bolted joint is Simple or Cylinder-head       : S
Total force on the joint                  (N) : 10000
Thickness of the upper part              (mm) : 16
Thickness of the lower part              (mm) : 16
Elastic modulus of the lower part       (GPa) : 80
Problem is to Check or to Design of Bolt      : C
```

Selection of Material

	Property class	Size range	Proof	Yield	Tensile	Material
			strength (MPa)			
1->	4.6	M5-M36	225	240	400	Low or Medium C steel
2->	4.8	M1.6-M16	310	340	420	Low or Medium C steel
3->	5.8	M5-M24	380	420	520	Low or Medium C steel
4->	8.8	M16-M36	600	660	830	Medium carbon steel
5->	9.8	M1.6-M16	650	720	900	Medium carbon steel
6->	10.9	M5-M36	830	940	1040	Low C Martensite steel
7->	12.9	M1.6-M36	970	1100	1220	Alloy steel

Material of the Bolt is Low or Medium Carbon Steel
Nominal Major Diameter of the Bolt (mm) : 8
Number of the bolt : 4
Preload on the bolt (N) : 0
It must be between 4941 and 7411.5 . Press any key.
Preload on the bolt (N) : 6000
The load is Static or Dynamic : S
Material of the members are Same or Not : S
There is any Gasket or Not : N

Tensile Force on the Bolt (N) : 10000
Thickness of the upper part (mm) : 16
Thickness of the lower part (mm) : 16
Elastic Modulus of the lower part (GPa) : 80
Problem is to Check or to Design of Bolt : C
Material of the Bolt is Low or Medium Carbon Steel
Nominal Major Diameter of the Bolt (mm) : 8
Number of the Bolt : 4
Preload on the Bolt (N) : 6000
The load is Static or Dynamic : S
Stiffness of the Bolt (MN/m) : 325.1548
Stiffness of the Members (MN/m) : 588.0589
Factor of Safety : 2.510844

Options Available

1-> To Input New Values
2-> To Change bolted joint type, Simple or Cylinder-head
3-> To Change upper part thickness
4-> To Change lower part thickness
5-> To Change lower part modulus of elasticity
6-> To Change problem type, Checking or Designing
7-> To Change loading type, Static or Dynamic
8-> To Change material of members as Same or Not
9-> To Change if there is any Gasket or Not
10-> To EXIT

Enter selection number : 6

Problem is to Check or to Design of Bolt : C

Selection of Material

Property class	Size range	Proof	Yield	Tensile	Material
		strength (MPa)			
1-> 4.6	M5-M36	225	240	400	Low or Medium C steel
2-> 4.8	M1.6-M16	310	340	420	Low or Medium C steel
3-> 5.8	M5-M24	380	420	520	Low or Medium C steel
4-> 8.8	M16-M36	600	660	830	Medium carbon steel
5-> 9.8	M1.6-M16	650	720	900	Medium carbon steel
6-> 10.9	M5-M36	830	940	1040	Low C Martensite steel
7-> 12.9	M1.6-M36	970	1100	1220	Alloy steel

Material of the Bolt is Low or Medium Carbon Steel
Nominal Major Diameter of the Bolt (mm) : 8
Number of the bolt : 4
Preload on the bolt (N) : 0
It must be between 6807.6 and 10211.4 . Press any key.
Preload on the bolt (N) : 8000

Tensile Force on the Bolt (N) : 10000
Thickness of the upper part (mm) : 16
Thickness of the lower part (mm) : 16
Elastic Modulus of the lower part (GPa) : 80
Problem is to Check or to Design of Bolt : C
Material of the Bolt is Low or Medium Carbon Steel
Nominal Major Diameter of the Bolt (mm) : 8
Number of the Bolt : 4
Preload on the Bolt (N) : 8000
The load is Static or Dynamic : S
Stiffness of the Bolt (MN/m) : 325.1548
Stiffness of the Members (MN/m) : 588.0589
Factor of Safety : 3.758963

Options Available

1-> To Input New Values
2-> To Change bolted joint type, Simple or Cylinder-head
3-> To Change upper part thickness
4-> To Change lower part thickness
5-> To Change lower part modulus of elasticity
6-> To Change problem type, Checking or Designing
7-> To Change loading type, Static or Dynamic
8-> To Change material of members as Same or Not
9-> To Change if there is any Gasket or Not
10-> To EXIT

Enter selection number : 10

Example 2: A pressure vessel is to be sealed using an asbestos gasket having a thickness of 3 mm and with a minimum seal pressure of 6 MPa. The vessel having inside, outside and mean diameters of 100, 210 and 160 mm respectively is to be rated for an internal pressure of 2 MPa. Members have same thickness of 15 mm and modulus of elasticity of 97 GPa. If a factor of safety of 1.5 is to be used, find the suitable bolt diameters and the number of bolts.

BOLT DESIGN

Bolted joint is Simple or Cylinder-head : C
Inside diameter of the cylinder (mm) : 100
Outside Diameter of the Cylinder (mm) : 210
Mean diameter between bolt hole center (mm) : 160
Working pressure in the cylinder (MPa) : 2
Thickness of the cylinder head (mm) : 15
Thickness of the cylinder (mm) : 15
Elastic modulus of the cylinder (GPa) : 97
Problem is to Check or to Design of Bolt : D
Factor of safety : 1.5
Pressure is Static or Dynamic : D
Reliability : 1>0.50 2>0.90 3>.95 4>.99 5>.999 6>.9999 7>.99999 : 2
Thread is Rolled or Cut or Fillet : R
Material of the members are Same or Not : S
There is any Gasket or Not : G
Minimum Gasket Seal Pressure (MPa) : 6

Selection of Gasket Material

	Material	Modulus of Elasticity (MPa)
1->	Cork	86
2->	Compressed asbestos	480
3->	Copper-asbestos	93000
4->	Plain rubber	69
5->	Spiral wound	280
6->	Teflon	240
7->	Vegetable fibber	120

The Thickness of Gasket (mm) : 3

Metric Class	Number of Bolt	Diameter (mm)	Required Preload-(N)	Torque (N.m)	Material
9.8	14	6	11701.5	14.04	Medium carbon steel
10.9	14	6	14352.8	17.22	Low-C. Martensite S.
10.9	15	6	14789.5	17.75	Low-C. Martensite S.

10.9	17	5	9393.0	9.39	Low-C. Martensite S.
12.9	17	5	11765.5	11.77	Alloy steel
10.9	18	5	9691.7	9.69	Low-C. Martensite S.
12.9	18	5	12074.3	12.07	Alloy steel
10.9	19	5	9958.8	9.96	Low-C. Martensite S.
12.9	19	5	12350.6	12.35	Alloy steel
10.9	20	5	10199.3	10.20	Low-C. Martensite S.
10.9	21	5	10416.8	10.42	Low-C. Martensite S.
12.9	24	4	6796.0	5.44	Alloy steel
12.9	25	4	6952.6	5.56	Alloy steel
12.9	26	4	7097.2	5.68	Alloy steel
12.9	27	4	7231.1	5.78	Alloy steel
12.9	28	4	7355.4	5.88	Alloy steel
12.9	29	4	7471.1	5.98	Alloy steel
12.9	30	4	7579.1	6.06	Alloy steel
12.9	31	3.5	5251.9	3.68	Alloy steel
9.8	32	4	5132.8	4.11	Medium carbon steel
12.9	32	3.5	5346.3	3.74	Alloy steel
12.9	33	3.5	5434.9	3.80	Alloy steel
12.9	34	3.5	5518.4	3.86	Alloy steel
12.9	35	3.5	5597.0	3.92	Alloy steel
12.9	36	3.5	5671.3	3.97	Alloy steel
12.9	37	3.5	5741.6	4.02	Alloy steel
12.9	38	3.5	5808.2	4.07	Alloy steel
12.9	39	3.5	5871.3	4.11	Alloy steel
9.8	41	3.5	3949.1	2.76	Medium carbon steel
12.9	42	3	3919.2	2.35	Alloy steel
12.9	43	3	3970.7	2.38	Alloy steel
12.9	44	3	4019.9	2.41	Alloy steel
12.9	45	3	4067.0	2.44	Alloy steel
12.9	46	3	4112.0	2.47	Alloy steel
12.9	47	3	4155.1	2.49	Alloy steel
12.9	48	3	4196.3	2.52	Alloy steel
12.9	49	3	4235.9	2.54	Alloy steel
12.9	50	3	4273.9	2.56	Alloy steel

Inside Diameter of the Cylinder (mm) : 100
Outside Diameter of the Cylinder (mm) : 210
Mean Diameter Between Bolt Hole Center (mm) : 160
Working Pressure in the Cylinder (MPa) : 2
Thickness of the cylinder head (mm) : 15
Thickness of the cylinder (mm) : 15
Elastic Modulus of the cylinder (GPa) : 97
Problem is to Check or to Design of Bolt : D
Factor of Safety : 1.5
Pressure is Static or Dynamic : D
Reliability Factor : .897
Thread is Rolled or Cut or Fillet : R
Material of the Gasket is Compressed asbestos
Minimum Gasket Seal Pressure (MPa) : 6
The Thickness of Gasket (mm) : 3
Stiffness of the Bolt (MN/m) : 44.3393

Stiffness of the Members (MN/m) : 3.149984

Options Available

1-> To Input New Values
2-> To Change bolted joint type, Simple or Cylinder-head
3-> To Change cylinder head thickness
4-> To Change cylinder thickness
5-> To Change cylinder modulus of elasticity
6-> To Change problem type, Checking or Designing
7-> To Change loading type, Static or Dynamic
8-> To Change material of members as Same or Not
9-> To Change if there is any Gasket or Not
10-> To EXIT

Enter selection number : 10

CHAPTER VI

COMPUTER AIDED DESIGN OF MECHANICAL SPRINGS

6.1. INTRODUCTION

This chapter is devoted to computer aided design of mechanical springs. Among many types of mechanical springs, in this thesis, the emphasis is given to helical compression springs.

Helical compression springs and their functions are briefly discussed in section 6.2. Section 6.3 is devoted to the analysis of springs in which the formulas for stresses and deflection are given. In section 6.4, design of helical compression springs are discussed. Computer aided design of springs is explained in section 6.5 and examples are presented in this section.

6.2 HELICAL COMPRESSION SPRINGS

Springs perform a wide variety of functions in mechanical systems. They can be employed to produce forces in mechanisms or to introduce considerable amounts of elastic deformation. They can also be used to absorb energy and to eliminate the effects of shock and vibration.

A typical helical compression spring is shown in Figure 6.1. For a specified material, its significant characteristics are uniquely defined by the wire diameter d, the mean coil diameter D, the active number of coils N, the end conditions which determine the number of inactive coils and its free length. The primary purpose as in the case of all mechanical elements, is to satisfy certain functional requirements of the machine that they are used in. Typical uses of helical compression springs may be summarized as follows:

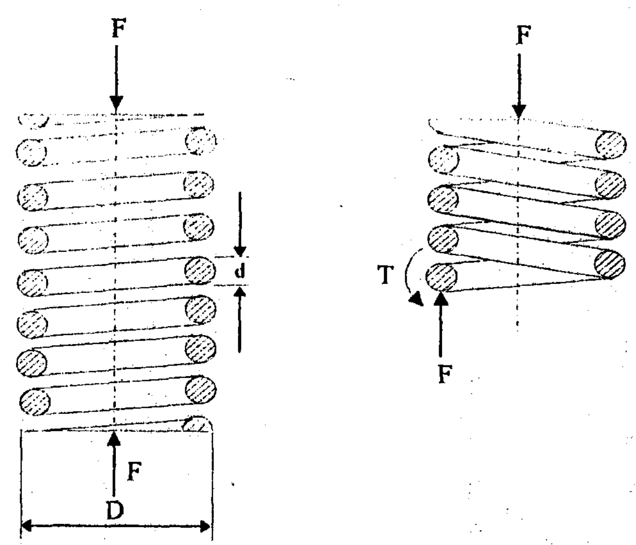


Figure 6.1 Axially Loaded Helical Compression Spring

a) Springs are used to control forces due to impact or shock loading and to control vibration. The energy of impact loading involves the product of a force, and a distance that corresponds to the deflection for a moderate load and therefore may be used to reduce the magnitude of transmitted force in shock loading. Examples of springs for this use are buffer springs for elevators and for car-track terminals, springs for railway cars and automobiles and springs in cushioned gears and sprockets.

b) Springs are employed to control motions and to apply forces to members. Examples of such uses are valve springs in internal-combustion engines, springs to produce pressure in brakes and clutches flexible supports for machinery or equipment.

c) Springs are used for storing energy, for example, in clocks, switch-throwing devices and starters.

d) Springs are employed to measure forces, as in scales.

The design of a new spring for an application involves the determination of a) the space that the spring is to fit, b) values of working forces and maximum deflection, c) environmental conditions such as corrosion etc., d) cost and quantities needed. By considering these requirements and working conditions, the designer will select suitable material and determine wire size, the number of active coils, the end conditions, the free length, the diameter of the coil and the spring stiffness to satisfy the deflection requirement. Since there are many factors that interact with each other, the design process is not simply introducing certain parameters into certain formulas but it is rather an iterative process which needs to be automated. There have been many attempts to automate the design of mechanical springs by using nomographs and charts [1-5]. In this thesis, an algorithm is developed and automated with a computer program which is interactive but requires little intervention of the user during the design process.

6.3 STRESSES AND DEFLECTIONS OF SPRINGS

Figure 6.1 shows a round wire helical compression spring loaded by the axial compression force F. Maximum shear stress in the wire may be computed using the equation;

$$\tau = \frac{Tr}{J} + \frac{F}{A} \tag{6.1}$$

Using the relations $T = FD/2$, $r = d/2$, $J = \pi d^4/32$ and $A = \pi d^2/4$ gives:

$$\tau = \frac{8FD}{\pi d^3} + \frac{4F}{\pi d^3} \tag{6.2}$$

The spring index which is a measure of coil curvature can be defined as:

$$C = \frac{D}{d} \tag{6.3}$$

where the mean spring diameter, D is equal to "D_o-d".

Substituting this relation into the Equation (6.2) and designating a shear stress multiplication factor K_S as below,

$$K_s = 1 + \frac{0.5}{C} \tag{6.4}$$

the following relation is obtained for the shear stress;

$$\tau = K_s \frac{8FD}{\pi d^3} \tag{6.5}$$

Equation (6.5) is quite general and applies for both static and dynamic loads. It gives the maximum shear stress in the wire. Introducing the curvature effect, the stress equation becomes;

$$\tau = K \frac{8FD}{\pi d^3} \tag{6.6}$$

where K is called Wahl correction factor. The value of K may be obtained from equation;

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C} \tag{6.7}$$

The curvature effect alone may be determined as;

$$K_c = \frac{K}{K_s} \tag{6.8}$$

For static loads curvature effect can be neglected, but for fatigue loads curvature effect is used as a fatigue-strength reduction factor. End condition or end type specifies the number of inactive coils (dead coils) which must be subtracted from the total number of coils. Active number of coils is determined as;

$$N = N_T - N_D \tag{6.9}$$

where N_D is the number of inactive coils. Spring constant (spring rate) is determined from the relation;

$$k = \frac{d^4 G}{8 D^3 N} \tag{6.10}$$

where G is the shear modulus of elasticity. Relationships between solid length, free length and maximum deflection can be written as;

$$L_s = N_T \ d \tag{6.11}$$

$$y_{max} = L_f - L_s \tag{6.12}$$

where L_s is equal to solid length, L_f is the free length of the spring and y_{max} is maximum deflection and may be expressed as;

$$\delta = y_{max} = \frac{F}{k} \tag{6.13}$$

Spring material is also important parameter in designing the spring. Shear yield strength, S_{sy}, can be determined from equations given in [1]:

$$S_{ut} = \frac{A}{d^m} \tag{6.14}$$

$$S_y = 0.75 S_{ut} \tag{6.15}$$

$$S_{sy} = 0.577 S_y \tag{6.16}$$

where A and m are constants specified for spring materials (Music wire, Oil tempered wire, Hard drawn wire, Chrome vanadium wire and Chrome silicon wire). These constants can be found in [1].

For critical frequency of helical spring, critical frequency, F_{RQ} is

$$F_{RQ} = \frac{1}{2}\sqrt{\frac{kg}{W}} \tag{6.17}$$

where k is the spring rate, g is the gravitational acceleration and W is equal to weight of the spring and can be calculated as;

$$W = \frac{\pi^2 d^2 D N \rho}{4} \tag{6.18}$$

and energy storage capacity U is determined as;

$$U = \frac{4F^2 D^3 N}{d^4 G} \tag{6.19}$$

6.4 DESIGN OF HELICAL COMPRESSION SPRINGS

In the design of compression springs, the loading condition must be carefully determined. If the load is of static type the process is rather an easy process compared to fatigue case. Depending upon the specified parameters, stress is calculated and compared with the strength of the spring material. In performing this, deflection requirements and space limitations, free length and solid length must always be taken into consideration.

Factor of safety, n_s, for static loading is determined as;

$$n_s = \frac{\tau}{S_{sy}} \tag{6.20}$$

In most applications, the load on the spring is subject to variations between certain limits. If this is the case, in addition to the check against maximum stress and deflection, fatigue failure must be taken into consideration. Spring design for fatigue applications involves more than simply selecting the proper cross section and spring style to fit the available space and carry the required loads. It also requires choosing an efficient and economic materials that satisfies the life requirements under given loads, stress and environmental conditions.

Unidirectional and reversed type of load would result in fatigue failure. The difference is that in the first case, stress is always applied in the same direction (compression). In the second case however, stress may take positive values. Most springs are subjected to unidirectional loading. If the spring is to be subjected to the load which varies between positive and negative values (reversed or repeated type), one must keep in mind that the spring must be preloaded in such a way that the resulting stresses will always be of compression type.

Forces usually ranges from minimum force F_{min} to a maximum force F_{max} then Equation (6.13) becomes

$$\delta_{max} = y_{max} = \frac{F_{max}}{k} \tag{6.21}$$

$$\delta_{min} = y_{min} = \frac{F_{min}}{k} \tag{6.22}$$

where y_{max} is the deflection corresponding to F_{max} and y_{min} is the deflection corresponding to F_{min}. If an eccentric loading case is considered, for an eccentric length of E_{ca}, F_{max} and F_{min} becomes

$$F_{max} = \frac{F_{max} D}{(D - 2E_{ca})} \tag{6.23}$$

$$F_{min} = \frac{F_{min} D}{(D - 2E_{ca})} \tag{6.24}$$

Alternating force F_a, and mean force F_m is calculated as

$$F_a = \frac{F_{max} - F_{min}}{2} \tag{6.25}$$

$$F_m = \frac{F_{max} + F_{min}}{2} \tag{6.26}$$

Then alternating shear stress, τ_a and mean shear stress, τ_m are:

$$\tau \quad K \frac{8F\ D}{\pi d} \tag{6.27}$$

$$\tau \quad K \frac{8F\ D}{\pi d} \tag{6.28}$$

Factor of safety, n_f for infinitive life requirement is determined from;

$$n = \frac{S}{\tau} \tag{6.29}$$

Factor of safety based on shear yield strength, just to check whether there is static failure or not, is determined from Equation (6.20).

In some applications, the design may be required to be based on finite life. In these cases, fatigue strength is used instead of endurance limit. Factor of safety for finite life is determined as :

$$n = \frac{S}{\tau} \tag{6.30}$$

6.5 COMPUTER AIDED DESIGN OF SPRINGS

6.5.1 Computer Program

The calculations involved in the design of springs are time consuming and laborious. For different applications and different requirements, there are so many considerations that must be taken into account, a sound design could be difficult to obtain with hand calculations.

A design algorithm is developed and it is automated with a computer program whose simplified flowchart is given in Figure 6.2. As can be seen from the flowchart, the program could be used for the calculation of factor of safety if all parameters are specified at the beginning or to select design parameters as to satisfy given requirements together with strength requirement.

As discussed in the previous sections, there are two basic loading conditions both may be used with the program. After data is input, the program directs the user to reach his goal for the specified problem.

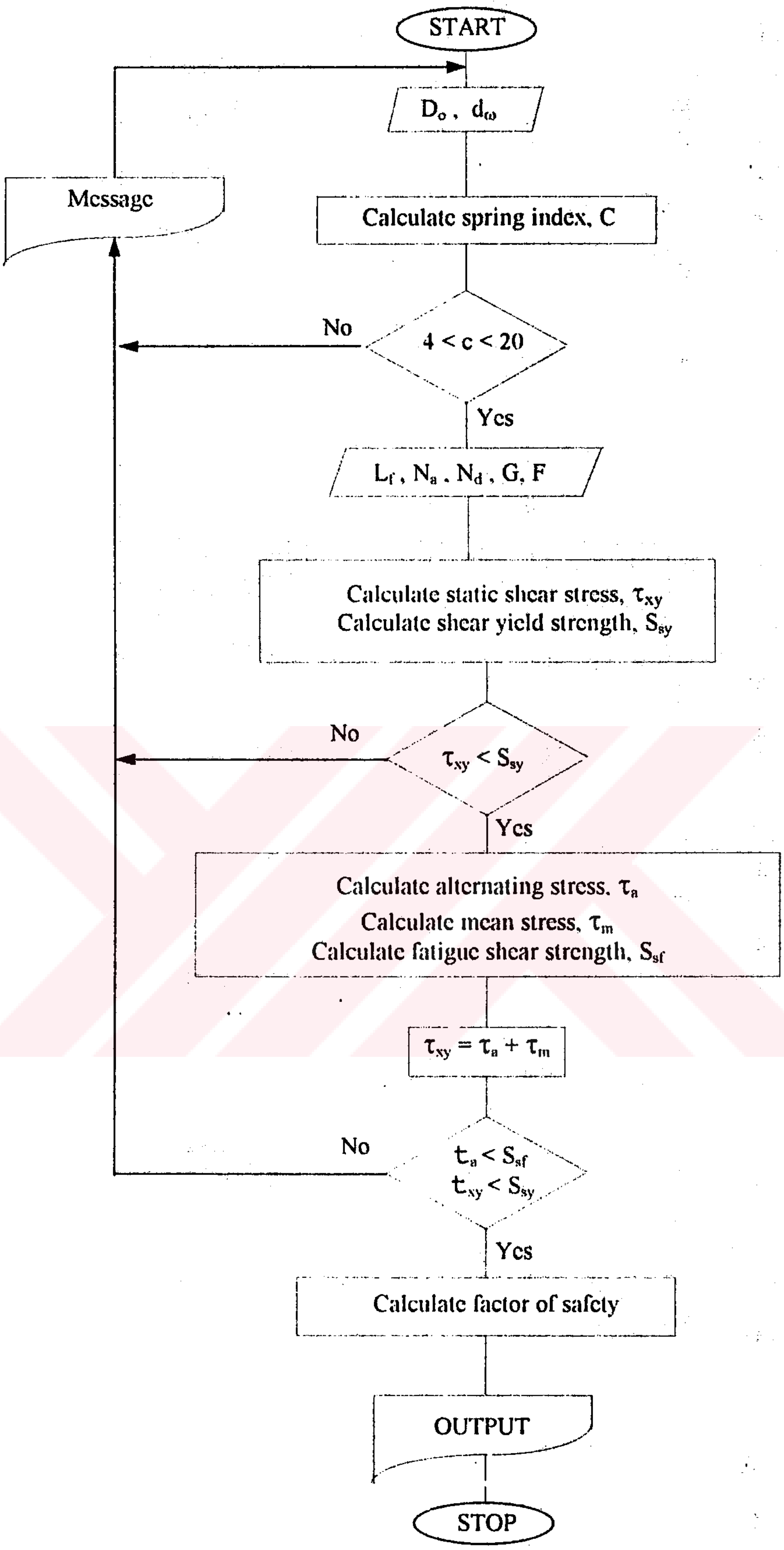


Figure 6.2 A Simplified Flowchart of the Program for Spring Design

6.5.2 Examples

The use of the program is verified through the examples for different applications. As will be appreciated, it is not possible to show all the capabilities of the program in this thesis due to space limitation. Two examples are prepared. The flow of the program is given for both of these examples.

Example 1: A 2.24 mm music wire spring has an outside diameter of 14.3 mm, a free length of 105 mm, 21 active coils and squared and ground ends. The spring is assembled with a preload of 45 N and will be operate to a maximum load of 225 N. Determine the factor of safety against a fatigue failure based on a life of 50000 cycles and 99 percent reliability.

HELICAL COMPRESSION SPRING DESIGN

Select Limitation
1-> Maximum Outside Diameter
2-> Minimum Inside Diameter
3-> Maximum Free Length
4-> Maximum Deflection
5-> Minimum Spring Rate
6-> Begin to Design

Limitation no : 6

Outside diameter (mm) : 14.3
Wire diameter (mm) : 2.24
Free length (mm) : 105
Number of active coils : 21

End Conditions
1-> Both ends plain
2-> Both ends squared
3-> Both ends squared and ground
4-> Both ends plain and ground

End conditions are Both ends squared and ground

Modulus of rigidity (GPa) : 79

Selection of Material

Material	Size range (mm)
1-> Music wire	0.10-6.5
2-> Oil-tempered wire	0.50-12
3-> Hard drawn wire	0.70-12
4-> Chrome vanadium	0.80-12
5-> Chrome silicon	1.60-10

Spring material is Music wire

Loading type is Static or Dynamic : D
The Deflection or Force is given : F
Forces are Eccentric or Not : N
Magnitude of maximum force (N) : 225
Magnitude of minimum force (N) : 45
Spring is Peened or Unpeened : U
Reliability : 1>0.50 2>0.90 3>0.95 4>0.99 5>.999 6>.9999 7>.99999 : 4
Spring life is Finite or Infinite : F
Number of cycles : 50000

Output Spring Specifications

Wire diameter	(mm) :	2.24
Outside diameter	(mm) :	14.3
Inside diameter	(mm) :	9.82
Free length	(mm) :	105
Solid length	(mm) :	51.52
Number of active coils	:	21
End conditions	:	Both ends squared and ground
Spring Material	:	Music wire
Spring index	:	5.384
Spring constant	(N/mm) :	6.749
Force makes spring solid	(N) :	360.960
Maximum force	(N) :	225.000
Maximum deflection	(mm) :	33.336
Critical frequency	(Hz) :	262.698
Energy storage capacity	(J) :	3.135
Static shear stress	(MPa) :	671.884
Shear yield strength	(MPa) :	834.758
Fatigue shear stress	(MPa) :	404.519
Factor of safety for static strength	:	1.242
Factor of safety for fatigue strength	:	1.505

Options Available

1-> To Input New Values
2-> To Change Limitations
3-> To Change Wire Diameter
4-> To Change Outside Diameter
5-> To Change Free Length
6-> To Change Spring Material
7-> To Change Number of Active Coils

8-> To Change End Conditions
9-> To Change Shear Modulus of Elasticity
10-> To Change Loading Type, Static or Fatigue
11-> To EXIT

Enter Selection Number : 11

Example 2: A valve spring is made of 9 mm chrome-vanadium steel wire, shot peened and both ends squared and ground. Inside is 75 mm, 8 active coils and free length is 180 mm. Solid length is 100 mm, length with valve is 150 mm and length when valve open is 130 mm. Determine the factor of safety.

HELICAL COMPRESSION SPRING DESIGN

Select Limitation
1-> Maximum Outside Diameter
2-> Minimum Inside Diameter
3-> Maximum Free Length
4-> Maximum Deflection
5-> Minimum Spring Rate
6-> Begin to Design

Limitation no : 6

Outside diameter (mm) : 93
Wire diameter (mm) : 9
Free length (mm) : 180
Number of active coils : 8

End Conditions
1-> Both ends plain
2-> Both ends squared
3-> Both ends squared and ground
4-> Both ends plain and ground

End conditions are Both ends squared and ground

Modulus of rigidity (GPa) : 79.3

Selection of Material

Material	Size range (mm)
1-> Music wire	0.10-6.5
2-> Oil-tempered wire	0.50-12
3-> Hard drawn wire	0.70-12

4-> Chrome vanadium 0.80-12
5-> Chrome silicon 1.60-10

Spring material is Chrome vanadium

Loading type is Static or Dynamic : D
The Deflection or Force is given : D
Value of Maximum Deflection (mm) : 50
Value of minimum deflection (mm) : 30
Spring is Peened or Unpeened : P
Reliability : 1>0.50 2>0.90 3>0.95 4>0.99 5>0.999 6>0.9999 7>0.99999 : 1
Spring life is Finite or Infinite : I

Output Spring Specifications

Wire diameter	(mm) :	9
Outside diameter	(mm) :	93
Inside diameter	(mm) :	75
Free length	(mm) :	180
Solid length	(mm) :	90
Number of active coils	:	8
End conditions	:	Both ends squared and ground
Spring Material	:	Chrome vanadium
Spring index	:	9.333
Spring constant	(N/mm) :	13.716
Force makes spring solid	(N) :	1234.434
Maximum force	(N) :	685.797
Maximum deflection	(mm) :	50.000
Critical frequency	(Hz) :	57.219
Energy storage capacity	(J) :	13.322
Static shear stress	(MPa) :	212.008
Shear yield strength	(MPa) :	599.665
Shear endurance limit	(MPa) :	423.837
Factor of safety for static strength	:	2.829
Factor of safety for fatigue strength	:	9.996

Options Available

1-> To Input New Values
2-> To Change Limitations
3-> To Change Wire Diameter
4-> To Change Outside Diameter
5-> To Change Free Length
6-> To Change Spring Material
7-> To Change Number of Active Coils
8-> To Change End Conditions
9-> To Change Shear Modulus of Elasticity
10-> To Change Loading Type, Static or Fatigue
11-> To EXIT

Enter Selection Number : 11

CHAPTER VII

COMPUTER AIDED DESIGN OF V-BELTS

7.1. INTRODUCTION

This chapter is devoted to the design of V-belt drive systems. V-belts are one of the most commonly used mechanical power transmission elements. The ability of transmitting higher torque at less width and tension cause them to be preferred to the other kind of flexible machine elements.

The drive system consists of driver and driven shaves and a belt wrapped around these shaves. These components are standardised by manufacturers, therefore this study is rather a selection process of these components. Design principles of V-belt systems can be found in any textbook on Mechanical Engineering Design [1-5]. Cross-sections of V-belts are standardised according to the type of applications and dimensions of the cross-sections and standard pitch lengths can be found in manufacturer's catalogues.

The power to be transmitted, shave diameters, centre distance, rotational speeds of the shafts, cross-section of belt are the important parameters considered in V-belt drive system. These parameters are not independent from each other, the procedure is an iterative procedure. A design method is developed and automated by means of a computer program. The design was based on the standardised and recommended limitations of manufacturer catalogues. When these limitations are not satisfied during the flow of program, the designer is warned to change of the values of certain parameters. The program enables the comparison of various solutions to the designer, and thus most suitable solution can be obtained for any specific application. Dynamic analysis of the system is also performed in the program. The use of program is explained by an example.

7.2. GENERAL PROPERTIES OF THE V-BELTS

A typical V-belt drive system is shown in Figure 7.1.

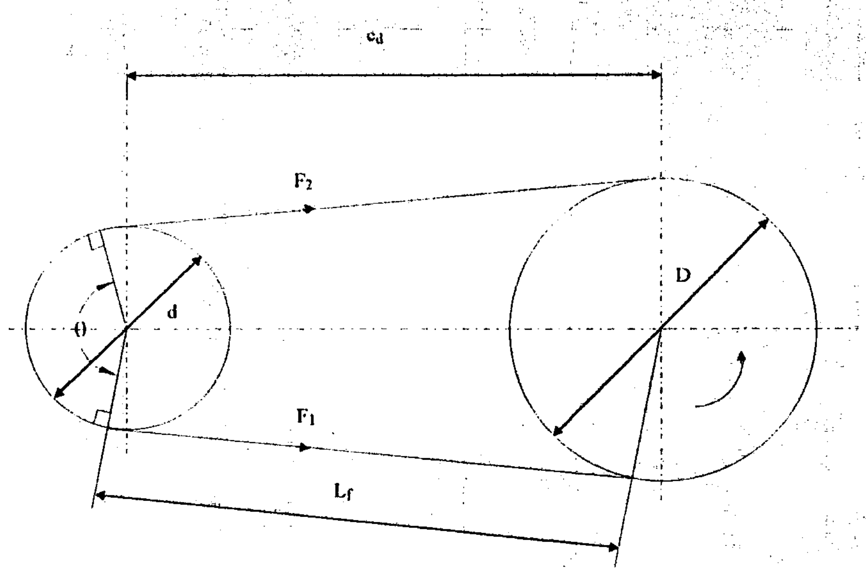


Figure 7.1 V-Belt Drive System

The design process is started with the calculation of the design power. The essential power for the design is the power required by the driving pulley. The effective factors on this power are the maximum load and the operating conditions. The design power for the belts is found by multiplying the driving motor power of the system and the service factor arising from the working conditions,

$$P_d = P \times S_f \tag{7.1}$$

where S_f is the service factor which is specified in manufacturer catalogues.

After the determination of the design power, the cross-section type of belt is selected according to the rotational speed of the small pulley and the design power. In manufacturer catalogues, the variation of the speed of the small-size pulley with respect to the design power is given for cross-section types of A, B, C, D, and E, where the load carrying capacity increases as going from A to E. After deciding the most proper cross-section type, the diameters of pulleys are selected according to the selected type of the belt. The diameters are selected as large as possible, because the life of the belt is considerably affected by pulley diameters [36].

Another important step in the design procedure is the calculation of pitch length of the selected belt. By using Figure 8.1, the pitch length of the belt is found as follows,

$$L_p = 2C_d + 1.57(D+d) + \frac{(D-d)^2}{4C_d} \tag{7.2}$$

where D and d are shave diameters and C_d is the centre distance.

Since excessive vibrations of the slack side of the belt will shorten the belt life, centre distances are recommended between diameter of larger shave and three times the sum of the shave diameters [37], that is,

$$D \leq C_d \leq 3(D + d) \tag{7.3}$$

The contact angle of the belt on a pulley is affective on the efficiency of the system. Because the belt is used in the transmission of either power or speed, the contact angle is being different than 180° on a large probability. The enveloping angle for small shave (small angle of contact) is calculated as follow;

$$\theta = \pi - 2\mathrm{Sin}^{-1}\frac{(D-d)}{2C_d} \tag{7.4}$$

Power is transmitted by the friction between pulley and the belt. Assuming friction is uniform throughout the arc of contact, the relation between the tight-side tension F_1 and the slack-side tension F_2 is;

$$\frac{F_1}{F_2} = e^{f\theta} \tag{7.5}$$

where θ is the contact angle and f is the coefficient of friction. In manufacturer catalogues, f is specified.

The power transmitted in watts is,

$$P = (F_1 - F_2)V \tag{7.6}$$

where V is the linear velocity of belt in meters per second. The velocity of belt is dependent on the working conditions and it was limited between 10 m/s and 25 m/s [37].

Centrifugal force is expressed as;

$$F_c = K_c\left(\frac{V}{1000}\right)^2 \tag{7.7}$$

and the bending force is found from;

$$F_b = \frac{K_b}{d} \tag{7.8}$$

where constants K_b and K_c are given in manufacturer catalogues, V is the linear velocity of belt in meters per second and d is shave diameter in millimetres.

The total peak forces on tight-side F_{T1} and slack-side F_{T2} are;

$$F_{T1} = F_1 + F_{b1} + F_c \tag{7.9}$$

$$F_{T2} = F_1 + F_{b2} + F_c \tag{7.10}$$

Another important factor for the system is the number of belt. The power to be transmitted becomes effective on the determination of the number of the belt. There is a certain capacity of each belt and the power that can be transmitted per belt is defined as rated power. The power-rating equation for a contact angle of 180^{0} and an average belt length is;

$$P_r = \left[C_1 - \frac{C_2}{d} - C_3(rd)^2 - C_4 \log(rd) \right](rd) + C_2 r\left(1 - \frac{1}{K_A}\right) \qquad (7.11)$$

where C_1, C_2, C_3 and C_4 are constants which depend upon belt section, r is the rpm of high-speed shaft divided by 1000, K_A is the speed ratio factor and d is the pitch diameter of small shave in inches for obtain P_r in horse-power or in millimetres for P_r in kilowatts. This power must be corrected for other contact angles and pitch lengths.

$$P_c = K_1 K_2 P_r \qquad (7.12)$$

where K_1 is the arc correction factor and K_2 is the belt length correction factor. The ratio of the design power calculated with equation (7.1) to the corrected rated power gives the total number of belts. However, the number of belt should not exceed 8 for the prevention of unbalanced power distribution on the belts [38].

The expected life in hours is determined from the ratio of equivalent peaks to belt passes per minute. Belt passes per minute is the ratio of linear velocity to pitch length of the belt. The equivalent peaks is calculated from;

$$\frac{1}{N_E} = \frac{1}{N_1} + \frac{1}{N_2} \tag{7.13}$$

where N_1 and N_2 are force peaks on small and large shave respectively. The force peaks on a shave is defined as;

$$N = \left(\frac{Q}{F_T}\right)^x \tag{7.14}$$

where Q and x are constants which depend upon belt section. However, the life of belt was recommended between 5000 and 25000 hours [39].

Natural frequency for flapping is based on the time required for a wave exited at one end of the belt free length to travel from one shave to the other and returning the starting point after reflection. If U is wave speed and if belt speed is small compared to U, the wave speed frequency is calculated with the following formula [40];

$$F_q = \frac{U}{2L_f} \tag{7.15}$$

where L_f is the free length of the belt and the parameter U is calculated from the equation [40];

$$U = \sqrt{\frac{F_t}{m}} \tag{7.16}$$

where F_t is the total tension force on the belt and m is the mass of belt per length.

The natural twist frequency is calculated as follows [40];

$$F_{qn} = \frac{1}{2\pi}\left(K_t \frac{J_1 + J_2(n_f/n_s)^2}{J_1 J_2 (n_f/n_s)^2}\right)^{\frac{1}{2}} \tag{7.17}$$

where K_t is calculated from;

$$K_t = \frac{d^2 AE}{2L_f} \tag{7.18}$$

The calculation of the values of J_1 and J_2 in equation (7.17) is not simple. The experimental and theoretical values of these parameters are not available in literature. In this work, the moments of inertia of the pulleys are calculated with a model in which the pulleys are divided in to certain number of slices which have cylindrical shape[40].

At the end of the dynamic analysis, the preload which must be applied statically on the system is calculated. The preload is applied to prevent the looseness for efficient operation of the system and it is determined as that;

$$F_s = \frac{F_1 + F_2}{2} + F_c \tag{7.19}$$

7.3. COMPUTER AIDED SELECTION OF V-BELTS

7.3.1 Computer Program

The calculations involved in selecting a V-belt in engineering design applications are complicated and laborious. A simplified flowchart for the selection procedure is given in Figure 7.2. Driving motor power, rotational speeds of driver and driven shaves and the service factor are the input to the program.

In the selection procedure, firstly, V-belts are selected so as to satisfy the linear velocity of the belt in the restricted range. Rated power must also be checked so as to be greater than zero. Finally, the belt life is asked if it is enough or not.

After these data are input, the program prints the belt cross-section types with minimum shave diameters and offers a suitable section types. After the user decides on the type, the program asks if the centre distance is fixed or not. If the centre distance is fixed, it is input to the program. This value is assigned as large shave diameter. If not, the small shave diameter is input, large shave diameter is calculated and the centre distance is assigned to the large shave diameter.

The iterative procedure goes on until the belt linear velocity is satisfied. When the constants which depend upon belt section is required, the program calls related tables.

After calculation of the design parameters, the program asks the user as if execute the program again by changing some parameters or exit the program.

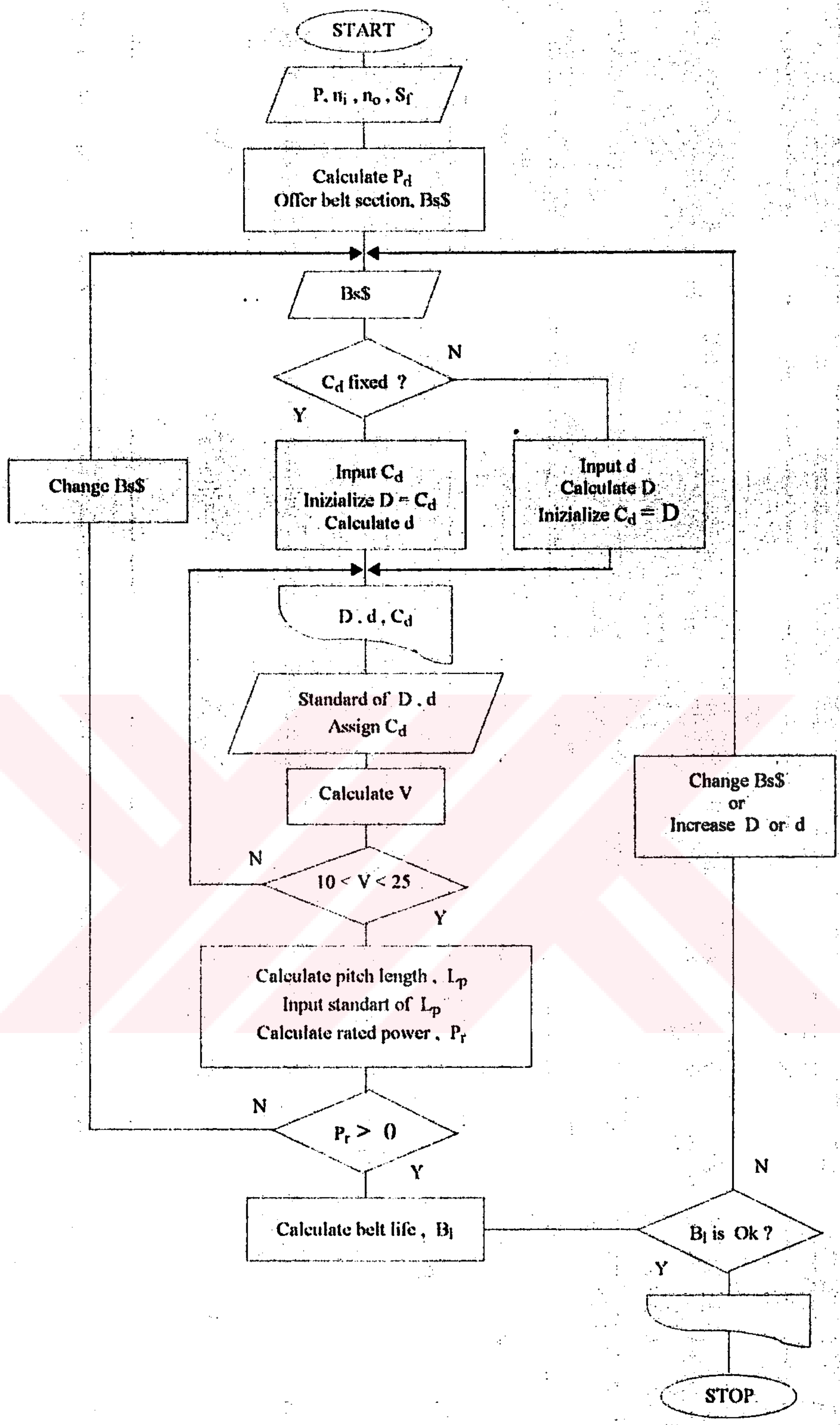


Figure 7.2 A Simplified Flowchart of the Selection Procedure for V-belts

7.3.2 Examples

In this work, the design of V-belt drive systems is automated with an interacting computer program.

By means of this program, the various alternatives are compared and the most proper result is found in the possible shortest time. All of the information about the V-belt drives are stored into the program, therefore the user is not necessarily to be a specialist on V-belt drive systems. A manufacturer catalogue which includes the standard belt lengths and the pulley diameters, is adequate for the user. This program also analyses the dynamic properties and the life of the system, and the loads acted upon the shaft in addition to the information to be obtained from the catalogues.

Example 1: It is required to select V-belt drive for a shewing press. Drive motor has a speed of 1440 rpm and power of 7.5 kW, while press has speed of 720 rpm, heavy working conditions about 16 hours a day and centre distance is restricted between 600 and 680 mm. The results are given as follows:

V-BELT DESIGN

Driver power (hp) : 10
Speed of the driver (rpm) : 1440
Speed of the driven (rpm) : 720

Service Factors For V-Belts

Agitators, paddle propeller	<1.0-1.2>	Line shafts..........	<1.2-1.6>
Bakery machinery...........	<1.0-1.2>	Machinery...........	<1.4-1.8>
Brick and clay	<1.2-1.6>	Machine tools........	<1.0-1.4>
Compressors................	<1.2-1.4>	Mills................	<1.4-1.6>
Compressors (ore sand).....	<1.3-1.6>	Oil field machinery..	<1.2-1.6>
Conveyors (packag)..........	<1.0-1.3>	Paper machinery.....	<1.2-1.6>

Crussing machinery..........	<1.4-1.6>	Printing machinery..	< 1.2 >
Fans and blowers............	<1.2-1.6>	Pumps(rotating)......	<1.2-1.4>
Generators and exiters......	< 1.2 >	Pumps(reciprocating).	<1.2-1.4>
Laundary machinery..........	< 1.2 >	Screens.............	<1.2-1.6>
Rubber plant machinery.....	<1.2-1.4>	Textile machinery....	<1.2-1.6>

Service factor : 1.3
B type belt is recommended.

Recommended minimum shave diameters for belt sections

Belt section	Minimum shave diameter (mm)
A	80
B	125
C	200
D	335
E	500
X	Not recommended

The belt section type is : B
Center distance is Fixed or Not : N
Driver shave diameter (mm) : 300
Driven shave diameter (mm) : 600.00
Driver shave diameter (mm) : 300.00
Center distance (mm) : 600.00

Standard value of driven shave diameter (mm) : 600
Standard value of driver shave diameter (mm) : 300
Recommended value of center distance (mm) : 640
Speed of the driven (rpm) : 720.00
Do you Agree with this speed or Not : A
Belt linear speed (m/s) : 22.62
Calculated pitch length of belt (mm) : 2728.16

Belt length	Correction factor	Belt length	Correction factor
960	0.81	2820	1.03
1040	0.83	2920	1.05
1090/1120	0.84	3130	1.06
1190	0.85	3330	1.07
1250	0.86	3530	1.09
1320	0.87	3740	1.10
1400	0.88	4090	1.11
1500	0.90	4200	1.13
1600	0.91	4480	1.14
1700	0.92	4650	1.15
1800	0.94	5040	1.16
1900	0.95	5300	1.18
1980	0.96	5760	1.19
2110	0.97	6140	1.23
2240	0.99	6520	1.24
2360	1.00	6910	1.25
2500	1.01	7290	1.26
2620	1.02	7670	1.27

Standard value of pitch length of belt (mm) : 2820

Driven shave diameter (mm) : 600.00
Driver shave diameter (mm) : 300.00
Pitch length of the belt (mm) : 2820.00
Center distance (mm) : 687.13
Do you want to Change them or Not : C

Standard value of driven shave diameter (mm) : 620
Standard value of driver shave diameter (mm) : 310
Recommended value of center distance (mm) : 650
Speed of the driven (rpm) : 720.00
Do you Agree with this speed or Not : A
Belt linear speed (m/s) : 23.37
Calculated pitch length of belt (mm) : 2797.06

Standard value of pitch length of belt (mm) : 2820

Driven shave diameter (mm) : 620.00
Driver shave diameter (mm) : 310.00
Pitch length of the belt (mm) : 2820.00
Center distance (mm) : 661.80
Do you want to Change them or Not : N
Effective belt length (mm) : 2820.00

Belt length	Correction factor	Belt length	Correction factor
960	0.81	2820	1.03
1040	0.83	2920	1.05
1090/1120	0.84	3130	1.06
1190	0.85	3330	1.07
1250	0.86	3530	1.09
1320	0.87	3740	1.10
1400	0.88	4090	1.11
1500	0.90	4200	1.13
1600	0.91	4480	1.14
1700	0.92	4650	1.15
1800	0.94	5040	1.16
1900	0.95	5300	1.18
1980	0.96	5760	1.19
2110	0.97	6140	1.23
2240	0.99	6520	1.24
2360	1.00	6910	1.25
2500	1.01	7290	1.26
2620	1.02	7670	1.27

Effective belt length correction factor : 1.03
Speed ratio : 2.000

Speed ratio range	Speed ratio factor
1.00-1.01	1.0000

1.02-1.04	1.0112
1.05-1.07	1.0226
1.08-1.10	1.0344
1.11-1.14	1.0463
1.15-1.20	1.0586
1.21-1.27	1.0711
1.28-1.36	1.0840
1.40-1.64	1.0972
Over 1.64	1.1106

Speed ratio factor : 1.1106
Arc of contact on small shave (Deg) : 152.91

Arc of contact on small shave	Correction factors	
	V to V	V to Flat
180	1.00	0.75
170	0.98	0.77
160	0.95	0.80
150	0.92	0.82
140	0.89	0.84
130	0.86	0.86
120	0.82	0.82
110	0.78	0.78
100	0.74	0.74
90	0.69	0.69

Arc correction factor : .93
Rated horse power (Hp) : 12.40
Calculated number of belts : 1.09
Number of belts : 2
Life of the belts (hour) : 974272.9
Do you Agree with this belt life or Not : A

Belt material is One of leather, rubber, cotton or Not : O

Material	Modulus of elasticity (MPa)
Leather	395
Cotton-woven	935
Rubber-covered	1325

Modulus of elasticity of belt (MPa) : 935
Driven pulley inertia ($kg.mm^2$) : 1973440.9
Do you want to Add additional inertia on driven shave or Not : N
Driver pulley inertia ($kg.mm^2$) : 168157.6
Do you want to Add additional inertia on driver shave or Not : N

Design constants and parameters

Centrifugal tension (N) : 90.91
Bending tension on small shave (N) : 210.02

Bending tension on large shave	(N) :	105.01
Tight side tension of belt	(N) :	278.40
Slack side tension of belt	(N) :	70.94
Peak force level on tight side	(N) :	579.32
Peak force level on slack side	(N) :	474.31
Force peaks on small shave	:	3.23E10
Force peaks on large shave	:	2.87E11
Equivalent force peaks	:	2.91E10
Belt passes per minute	(1/min) :	497.31
Arc of contact on small shave	(deg) :	152.91
Arc correction factor	(K1) :	0.93
Speed ratio factor	(Ka) :	1.11
Belt length correction factor	(K2) :	1.03

Rated power constants

C1 = .05185

C2 = 2.273

C3 = 1.76E-08

C4 = .00793

Design constants

Bending force constant	(Kb) :	576.00
Centrifugal force constant	(Kc) :	0.96
Life calculation constant	(X) :	10.92
Life calculation constant	(Q) :	1193.00

Free length of belt	(mm) :	643.39
Belt linear speed	(m/s) :	23.37
Mass of belt per length	(kg/m) :	0.19
Belt section area	(mm^2) :	148.00
Belt modulus of elasticity	(MPa) :	935.00
Equivalent torsional stiffness	:	1053470.8

Final specification of the drive design

Power of the drive	(hp) :	10.00
Service factor	:	1.30
Number of belts	:	2.00
Belt section	:	B
Driver shave diameter	(mm) :	310.00
Driven shave diameter	(mm) :	620.00
Center distance	(mm) :	661.80
Effective belt length	(mm) :	2820.00
Driver shave speed	(rpm) :	1440.00
Driven shave speed	(rpm) :	720.00
Life of the belt	(hour) :	974272.88
Flapping frequency on tight side	(Hz) :	34.05
Flapping frequency on slack side	(Hz) :	22.54
System natural frequency	(Hz) :	39.87
Total radial load on shave	(N) :	1037.32
Required static pretension	(N) :	265.58

Options Available
1-> To Input New Values
2-> To Change Service Factor
3-> To Change Driver Power
4-> To Change Driver Speed
5-> To Change Driven Speed
6-> To Change Belt Section
7-> To Change Driver Shave Diameter
8-> To Change Driven Shave Diameter
9-> To Change Center Distance
10-> To Change Belt Length
11-> To Change Number of Belts
12-> To EXIT

Enter selection number : 6

Recommended minimum shave diameters for belt sections

Belt section	Minimum shave diameter (mm)
A	80
B	125
C	200
D	335
E	500
X	Not recommended

The belt section type is : C
C section is allowed only for automotive industry.
Do you prefer to Change belt section or Not : N

Driver shave diameter (mm) : 250
Driven shave diameter (mm) : 500.00
Driver shave diameter (mm) : 250.00
Center distance (mm) : 500.00

Standard value of driven shave diameter (mm) : 500
Standard value of driver shave diameter (mm) : 250
Recommended value of center distance (mm) : 640
Speed of the driven (rpm) : 720.00
Do you agree with this speed or Not : A
Belt linear speed (m/s) : 18.85
Calculated pitch length of belt (mm) : 2481.91

Belt length	Correction factor	Belt length	Correction factor
1400	0.89	4120	1.09
1500	0.90	4220	1.10
1630	0.92	4500	1.11
1830	0.94	4680	1.13
1900	0.94	5060	1.14
2000	0.95	5440	1.13
2160	0.96	5770	1.14
2260	0.98	6150	1.13

2390	0.99	6540	1.14
2540	1.00	6920	1.13
2650	1.01	7300	1.14
2800	1.02	7680	1.13
3030	1.03	8060	1.14
3150	1.04	8440	1.13
3350	1.06	8820	1.14
3550	1.07	9200	1.13
3760	1.08		

Standard value of pitch length of belt (mm) : 2540

Driven shave diameter (mm) : 500.00
Driver shave diameter (mm) : 250.00
Pitch length of the belt (mm) : 2540.00
Center distance (mm) : 669.58
Do you want to Change them or Not : N
Effective belt length (mm) : 2540.00

Belt length	Correction factor	Belt length	Correction factor
1400	0.89	4120	1.09
1500	0.90	4220	1.10
1630	0.92	4500	1.11
1830	0.94	4680	1.13
1900	0.94	5060	1.14
2000	0.95	5440	1.13
2160	0.96	5770	1.14
2260	0.98	6150	1.13
2390	0.99	6540	1.14
2540	1.00	6920	1.13
2650	1.01	7300	1.14
2800	1.02	7680	1.13
3030	1.03	8060	1.14
3150	1.04	8440	1.13
3350	1.06	8820	1.14
3550	1.07	9200	1.13
3760	1.08		

Effective belt length correction factor : 1
Speed ratio : 2.000

Speed ratio range	Speed ratio factor
1.00-1.01	1.0000
1.02-1.04	1.0112
1.05-1.07	1.0226
1.08-1.10	1.0344
1.11-1.14	1.0463
1.15-1.20	1.0586
1.21-1.27	1.0711
1.28-1.36	1.0840
1.40-1.64	1.0972

Over 1.64 1.1106

Speed ratio factor : 1.1106
Arc of contact on small shave (deg) : 158.48

Arc of contact on small shave	Correction factors V to V	V to Flat
180	1.00	0.75
170	0.98	0.77
160	0.95	0.80
150	0.92	0.82
140	0.89	0.84
130	0.86	0.86
120	0.82	0.82
110	0.78	0.78
100	0.74	0.74
90	0.69	0.69

Arc correction factor : .945
Rated horse power (hp) : 20.75
Calculated number of belts : 0.66
Number of belts : 1
Life of the belts (hour) : 18200.5
Do you Agree with this belt life or Not : N

If the life is less than expected :
Increase shave diameters or number of belts. Change belt section.

Final specification of the drive design

Power of the drive	(hp) :	10.00
Service factor	:	1.30
Number of belts	:	1.00
Belt section	:	C
Driver shave diameter	(mm) :	250.00
Driven shave diameter	(mm) :	500.00
Center distance	(mm) :	669.58
Effective belt length	(mm) :	2540.00
Driver shave speed	(rpm) :	1440.00
Driven shave speed	(rpm) :	720.00
Life of the belt	(hour) :	18200.54
Flapping frequency on tight side	(Hz) :	34.05
Flapping frequency on slack side	(Hz) :	22.54
System natural frequency	(Hz) :	39.87
Total radial load on shave	(N) :	1039.97
Required static pretension	(N) :	265.58

Options Available
1-> To Input New Values
2-> To Change Service Factor
3-> To Change Driver Power

4-> To Change Driver Speed
5-> To Change Driven Speed
6-> To Change Belt Section
7-> To Change Driver Shave Diameter
8-> To Change Driven Shave Diameter
9-> To Change Center Distance
10-> To Change Belt Length
11-> To Change Number of Belts
12-> To EXIT

Enter selection number : 11

Number of belts : 1
New number of belts : 2
Rated horse power (hp) : 20.75
Life of the belts (hour) : 324557.6
Do you Agree with this belt life or Not : A

Design constants and parameters

Centrifugal tension	(N) :	105.14
Bending tension on small shave	(N) :	723.39
Bending tension on large shave	(N) :	361.70
Tight side tension of belt	(N) :	339.57
Slack side tension of belt	(N) :	82.32
Peak force level on tight side	(N) :	1168.10
Peak force level on slack side	(N) :	806.40
Force peaks on small shave	:	8.81E09
Force peaks on large shave	:	5.53E11
Equivalent force peaks	:	8.67E09
Belt passes per minute	(1/min) :	445.27
Arc of contact on small shave	(deg) :	158.48
Arc correction factor	(K1) :	0.94
Speed ratio factor	(Ka) :	1.11
Belt length correction factor	(K2) :	1.00

Rated power constants

C1 = .1002
C2 = 4.04
C3 = 3.33E-08
C4 = .015

Design constants

Bending force constant	(Kb) :	1600.00
Centrifugal force constant	(Kc) :	1.72
Life calculation constant	(X) :	11.17
Life calculation constant	(Q) :	2038.00

Free length of belt	(mm) :	657.81
Belt linear speed	(m/s) :	18.85
Mass of belt per length	(kg/m) :	0.32

Belt section area	(mm^2) :	244.00
Belt modulus of elasticity	(MPa) :	935.00
Equivalent torsional stiffness	:	670120.9

Final specification of the drive design

Power of the drive	(hp) :	10.00
Service factor	:	1.30
Number of belts	:	2.00
Belt section	:	C
Driver shave diameter	(mm) :	250.00
Driven shave diameter	(mm) :	500.00
Center distance	(mm) :	669.58
Effective belt length	(mm) :	2540.00
Driver shave speed	(rpm) :	1440.00
Driven shave speed	(rpm) :	720.00
Life of the belt	(hour) :	324557.59
Flapping frequency on tight side	(Hz) :	36.54
Flapping frequency on slack side	(Hz) :	23.73
System natural frequency	(Hz) :	31.80
Total radial load on shave	(N) :	1245.81
Required static pretension	(N) :	316.08

Options Available
1-> To Input New Values
2-> To Change Service Factor
3-> To Change Driver Power
4-> To Change Driver Speed
5-> To Change Driven Speed
6-> To Change Belt Section
7-> To Change Driver Shave Diameter
8-> To Change Driven Shave Diameter
9-> To Change Center Distance
10-> To Change Belt Length
11-> To Change Number of Belts
12-> To EXIT

Enter selection number : 12

CHAPTER VIII

COMPUTER AIDED DESIGN OF BRAKES

8.1 INTRODUCTION

This chapter is devoted to computer aided design of brakes. Brakes are discussed in section 8.2. General braking requirements are presented in section 8.3. Braking system parameters considered most important are discussed in 8.4. Section 8.5 is devoted to Computer aided design of braking systems and the use of the computer program is illustrated with examples in this section.

8.2 BRAKES

Brakes are devices, which transform the energy of motion to the other forms of energy in order to bring the body to rest. Brakes are usually used to slow , stop or to allow its motion by a constant speed, by transforming its energy of motion into heat and dissipating it from braking mechanism. The speed of transforming governs the rate of retardation of the body.

If a brake is used on a moving body, it brings about a decrease of kinetic energy E_k or it opposes a loss of potential energy E_p, or both. The first consequence is an increase in the inter-molecular energy of the contacting bodies, mostly near the contacting surfaces. That is, the temperature of the bodies increases which results in the frictional work being eventually dissipated to the surroundings as heat. For this reason the capacity of a brake is often stated in terms of the amount of frictional work which is absorbed in a particular time. Sometimes, a brake has to be applied steadily over long periods of time, in which case it must be able to radiate and convect the heat to atmosphere at such a rate that the steady-state temperature is below a damaging value. Sometimes, the brake is applied for short intervals of time (intermittently) with enough time between applications for it to cool to a value close to the environmental temperature in which case it may safely absorb energy at a much higher rate.

The effectiveness of the brake may greatly decrease shortly after it begins to act continuously, a phenomena called "fade". This is basically due to a significant decrease in the coefficient of friction at the high surface temperatures induced.

The permissible normal pressures between the braking surfaces depend in varying degree on the brake-lining material, the coefficient of friction and on the maximum rate at which energy is to be absorbed. The higher the pressure, the greater is the rate of wear and the energy absorbed at a particular speed.

If a moving vehicle is considered, it processes energy of motion which must be transformed into some other form of energy of motion in order to bring the vehicle to rest. The speed of transformation governs the rate of retardation of the vehicle.

In the case of machines, brakes are used to (1) to slow or stop the machines by transforming its energy of motion into heat and (2) to control the speed of a descending load.

The usage area of brakes is very wide and there are many types of brakes. In this thesis, only the following mechanical brakes are studied.

1. Rim type with internally expanding shoes
2. Rim type with externally contracting shoes
3. Band types
4. Disc types

Schematic representations of these brake systems are given in Figures 8.1 to 8.4 respectively.

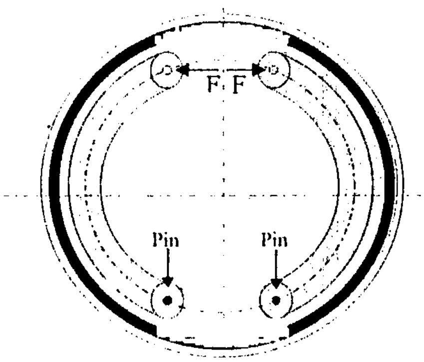


Figure 8.1. Rim Type of Brake with Internally Expanding Shoes

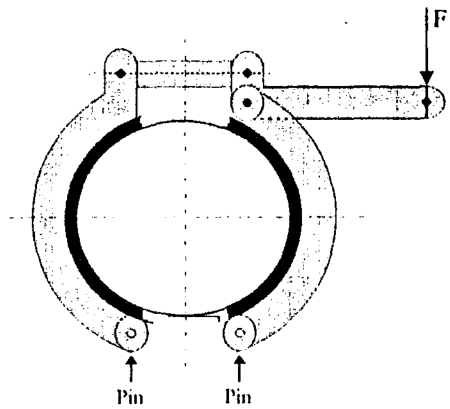


Figure 8.2 Rim Type of Brake with Externally Contracting Shoes

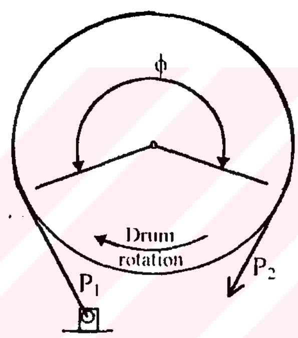


Figure 8.3 Band Type of Brake

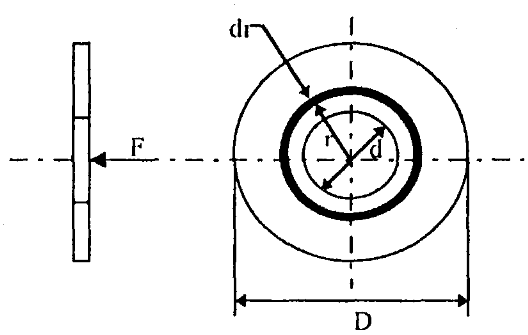


Figure 8.4 Disc Type of Brake

8.3 GENERAL BRAKING REQUIREMENTS

Brakes found applications in vehicles and they are considered as the most important and essential units of vehicles. The principles to be discussed here are to apply vehicles and they can also be applied for other applications.

The brake should be capable of stopping the vehicle within the shortest distance possible or decelerating it at the maximum possible rate. The braking system must be completely reliable and not be affected by temperature, water, road grit or dust, nor should its performance deteriorate if its components wear. The brakes and brake system should require as little maintenance and adjustment as possible.

The deceleration produced should ideally be uniform throughout an application. The response time of the whole system, should be as small as possible. The brakes should be free from vibration and from noise such as sequel.

In the applications of brakes on the vehicles, the deceleration always be same pedal effort exerted by the driver, increase with pedal effort in a regular way, and not require induce effort. The vehicle should not deviate to right or left as a result of braking and the brakes must be not interact with the suspension or steering. When a wheel is locked, the direction of the frictional force is independent of the orientation of the wheel and so cannot be used to control the direction of the vehicle. If the front wheels lock the front of the vehicle tends to slide forward in a straight line, the vehicle as a whole being restrained by the rear wheels. If all four wheels lock simultaneously, the vehicle will slide down the member out of control.

The loads carried by front and rear wheels of a vehicle are not in general equal, and so to utilise the adhesion more efficiently the designer can arrange that the braking effort is divided between front and rear wheels in the same ratio as the total weight of the vehicle is divided between front and rear axles.

Modern braking systems employ either a rotating drum or disc, the necessary friction is obtained by pressing a stationary shoe or pad against the rotating member. The operating system for brake can be (1) mechanical, (2) hydraulic, or (3) pneumatic (air). Brake fluid should have a low freezing point, high boiling point, and low viscosity (thin). The characteristics of brake fluid must not injure the rubber seal. The vegetable-based oil is preferred as brake fluid.

8.4 PARAMETERS OF BRAKING SYSTEMS

In analysing and designing of the braking systems, the following parameters are considered to be the most important, since these parameters determine the type and efficiency of the brake. They are listed as;

a. The actuating force.

b. The torque transmitted.

c. The energy loss.

d. The temperature rise.

a) The actuating force: The forces acting on the pads or friction materials by the piston are the actuating force. When the actuating systems comes into contact with the rotating surface, the actuating system applies an actuating force on the rotating drum or disc proportional to the pressure in the actuating system. It is generated at the hydraulic or mechanic brake actuating systems.

In the case of big electrical machines the rotating disc is braked by the magnetic field force. In the case of vehicles, the actuating force is proportional to the pressure in the hydraulic actuating system oil to the pistons.

b) Torque: The torque transmitted is related to the actuating force, the coefficient of friction, and the geometry of the brake. Frictional torque is important because the braking capacity is usually expressed in terms of frictional torque's. If T_f is constant or assumed to be some average value, the work of a frictional force on the surface of a rotating drum of diameter D, the energy in braking, can be expressed as:

$$U = 2\pi n T_f \quad \text{or} \quad T_f = U / (2\pi n) \qquad (8.1)$$

where n is the rotational speed of the drum while F and T_f are acting. For a constant angular acceleration α, frictional torque is determined from $T_f = I*\alpha$ where I is the moment of the inertia.

Sometimes a brake is to change the speed to a state of rest and the amount of frictional horsepower is calculated from:

$$P_f = T_f\,\omega/746 \qquad (8.2)$$

c) Energy Loss: The brake absorbs energy of motion and dissipates heat to the surrounding by air flow. The absorbed energies may be in different types and in different severity.

In nature, there are two well known types of energy. They are kinetic and potential energies. In a system, if there is a displacement between any two points which have different world coordinates from earth to elevated other points, the potential energy must be take into account. If there is a linear or rotational movement is concerned we must deal with the kinetic energy. The types of energy also affects the braking capacity.

The limiting capacity of a brake is commonly expressed in terms of the maximum instantaneous rate of energy absorption (P_f).

In elevation: The body being braked may undergo a change of potential energy. $E_k = W\,(h_1\text{-}h_2)$, where a body which has weight of W moves from an elevation h_1 to h_2 each measured in meter.

In translation: The change of kinetic energy of a translating body of weight W is $E_k = (\,W/2g)\,(V_i^2 - V_f^2)$, where g=9.81 m/ s^2.

In rotation: The change in kinetic energy is expressed as $E_k = (I/2)(\omega_i^2-\omega_f^2)$, where I is the moment of inertia of the body about its axis of rotation.

In rolling: The change in kinetic energy is expressed as $E_k = (I/2)(\omega_i^2 - \omega_f^2) + (W/2g)(V_i^2 - V_f^2)$.

The total energy that the brake absorbs is the decrease of the stored mechanical energy brought about the braking. This energy is dissipated as heat to the surrounding. Due to complexity of the mechanism of heat generation, it is not possible to determine the actual temperature attained by the rim and it is also difficult to compute time required for brake to cool. But it is possible the calculation of the rate of heat loss at a given temperature since heat loss must be equal to the rate of heat generation.

d) Temperature rise: The heating affects are naturally quite complex and local temperatures at points of contact may be very high. The part of the total frictional energy that is stored in braking elements such as drum or disc, has been virtually estimated as %75 or up. For very short applications of brake, say a few seconds, the percentage may increase up to % 100 at the instant that the brake is released.

For peak short time requirements, it is generally assumed that all frictional energy is absorbed by the adjacent metal in the drum wheel and temperature increase of the metal is calculated as:

$$\Delta T = U / m\,C \qquad (8.3)$$

where m and C are the mass and the specific heat of the metal.

8.5 COMPUTER AIDED DESIGN OF BRAKING SYSTEM

8.5.1 Computer Program

The calculations involved in the design of braking systems are complicated and laborious. As noticed in the text above, formulations are not given for most of the calculations. Formulas from textbook by Nieman [3] are adopted for vehicle applications and formulas from textbook by Shigley [1] are adopted for general applications. Due to space limitations, these formulations are not given in thesis, one may refer to these references for more information.

Considering the types of applications and configurations, the computer program is prepared on modular basis. That is, different modules are applicable for different types and applications.

8.5.2 Examples

The design of braking systems is verified through the examples by using computer program. In this study, firstly user is to decide on the type of braking system to be employed for the given mechanism. By asking questions, the program will direct the user to reach to a sound solution for his design problem. A database for the braking material and for other relevant design parameters is prepared to ease the design procedure.

All the possibilities are concerned in the computer program. By using this computer program , one can design every type of brake which are mentioned in section 8.3. The use of the program and its modules practical examples are prepared and the flow of the program for these examples are presented below.

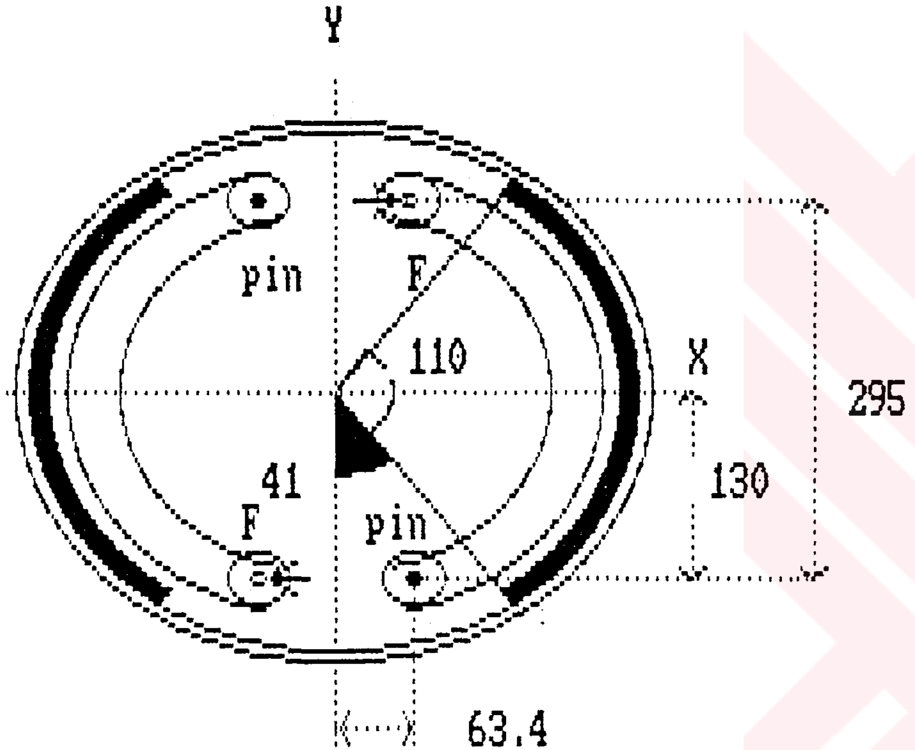
Example 1: A two-shoe truck brake whose dimensions are given has a coefficient of friction of 0.3 and is to have a maximum pressure of 0.689 MPa on each shoe. Determine the actuating force and the braking torque for clock wise rotation.

SIMPLE BRAKE DESIGN

```
Type of the brake              : Disk or druM : Drum
Type of the drum brake         : Internal, External or Band : Internal
Number of shoe pair            : 1
Pins are Crossed or Not        : C
Diameter of Drum       (mm) :  400
Maximum pressure      (MPa) :  .689
Coefficient of friction     :  .3
Face width of pad      (mm) :  125
Direction of rotation          . cW or Ccw : Cw

Application Field of Brake            : Drum
Type of the Brake                     : Internal
Number of Shoe Pair is                :    1
Angle of the Project the Lining       :  110
Angle between Lining end and Pin      :   41
The friction Coefficient is           :   .3
Diameter of drum is             (mm) :   400
The width of friction Surface (mm) :   125
Maximum pressure               (MPa) :  .689
Torque capacity of Brake       (N.m) :  3182.189
Actuating force                  (N) :  6518.261
Reaction force                   (N) :  18972.23

Do you want to Continue or Not   : N
```



INTERNAL BRAKE BOTH-NON SELF E.

EXPLAINATION

If the drum rotates CW both shoes are self energizing.

Otherwise, they are both non self energizing

Width of pad
b (mm) :125.00
Actuating force
F (kN) : 0.000

Press any key to continue

Example 2: A crane which has external contracting brake carries a load of 250 kg with a speed of 5 m/s. If the drum rotates with a speed of 300 rpm and the efficiency of mechanism is 90 percent, select a suitable material having a thickness of 3 mm.

BRAKE DESIGN

Options : 1-> Car 2-> Truck 3-> Crane 4-> Hoist 5-> Elevator
Application Field of Brake : Crane
Type of the Brake is External or Band : External
Velocity of load (m/s) : 5
Rotational speed of drum (rpm) : 300
Mass Moment of Inertia (kgm^2) : 50
Amount of Load (kg) : 250
Efficiency of Mechanism (%) : 90

Type	b/d	KU (kg/mm^2)	A (m/s^2)
Crane brake	0.3-0.4	0.002-0.008	Load deceleration, a < 1.4
Keeping brake	0.3-0.4	0.0075	
Stoping Brake	0.3-0.4	0.002-0.004	Braking time, Td : 0.5-5 sec
Lowering Brake	0.3-0.4	0.0025	

The b/d ratio : .35
The constant KU : .004
Duration time per braking (sec) : 1
Number of braking per hour : 60
Thickness of friction material (mm) : 3
Ratio of Effective area to Available area (0.1-0.99) : .5
Direction of load is Up or Down : D

Friction material	Friction coefficient	Temperature	Pressure
1> Synthetic reinous cotton web	0.4-0.6	150	1.17
2> Fenol-Synthetic reinous	0.25	150	0.68
3> Synthetic reinous asbestos web	0.3-0.5	300	1.92
4> Asbestos synthetic reinous with hydraulic pressing	0.2-0.35	500	7.84

Select friction material : Fenol-sr
Coefficient of friction : .25
Surrounding temperature : 25
Wear ratio (125-200) : 160

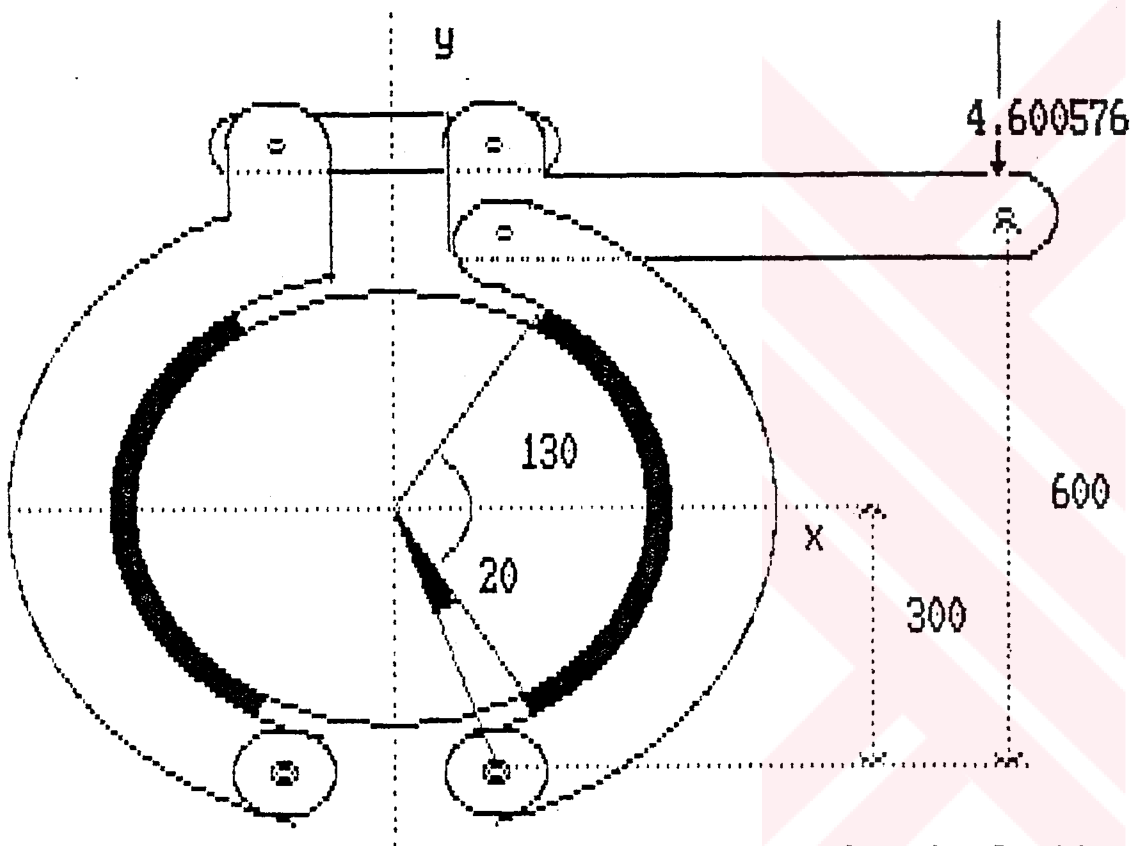
Application field of brake : Crane
Type of the brake : External
Velocity of load (m/s) : 5
Rotational speed of drum (rpm) : 300
Number of braking per hour : 60

The amount of load (kg) : 250
Efficiency of running mechanism (%) : 90
Angle of project lining (Deg) : 130
Angle between lining end and pin : 20
The friction material is : Fenol-sr
The friction coefficient is : . 25
The surrounding temperature (C) : 25
Thickness of friction material (mm) : 3
The application duration (sec) : .5
Direction of the drum : Down
The wear ratio : 160
The b/d ratio : .35
The constant KU : .004

Diameter of drum (mm) : 409
The width of friction surface (mm) : 143
Torque capacity of brake (kN.m) : .9408178
Maximum temperature (C) : 109.321
Maximum pressure (MPa) : .1885587
Actuating force-1 (kN) : 4.600576
Actuating force-2 (kN) : 4.600576
Total actuating force (kN) : 9.201153
To decrease actuating force increase diameter and friction coefficient
Braking time for lowering load (s) : 1
Braking distance, lowering load (m) : 2.5
In order to decrease the braking distance, increase deceleration
Length of friction surface (mm) : 927.8165
Life of friction material (w.hour) : 30100.48
In order to increase life of friction material
decrease wear ratio, increase friction material thickness and surface
Power (HP) : .1652935
Maximum normal force (kN) : 9.947191
Maximum friction force (kN) : 2.486798

Do You want to Continue or Not : N

EXTERNAL BRAKE TWO SHOES



REMARK !

Distance between friction material and frame is called as C3 (mm) = 90.00

Diameter of drum D (mm) =409.000

Thickness of friction material Tf (mm) = 3.000

Press any key to Continue

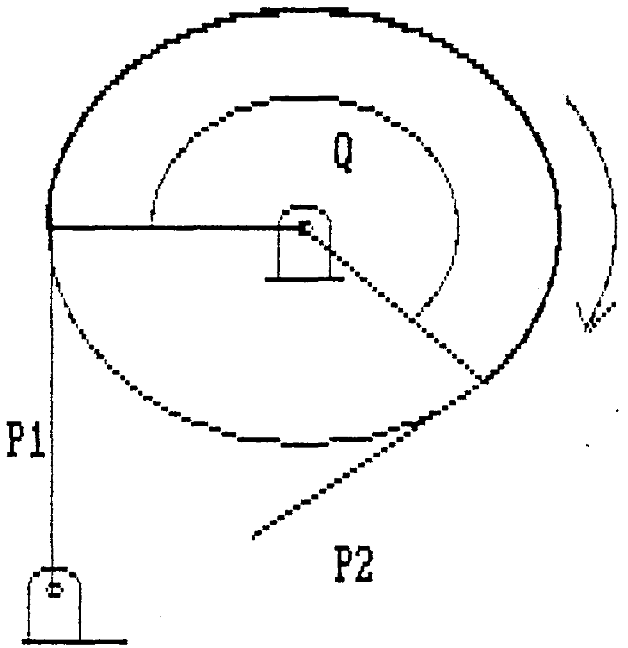
Example 3: A simple band brake has a coefficient of friction of 0.3 mm and is to have a maximum pressure of 1 MPa. If the drum diameter is 300 mm and the contact angle is 270 ^{0}, determine the braking torque.

SIMPLE BRAKE DESIGN

Type of the brake : Disk or druM : Drum
Type of the drum brake : Internal, External or Band : Band
Diameter of drum (mm) : 300
Maximum pressure (MPa) : 1
Coefficient of friction : .3
Direction of rotation : cW or Ccw : Cw

Application Field of Brake : Drum
Type of the Brake : Band
Angle of the Project the Lining : 270
The friction Coefficient is : .3
Diameter of drum is (mm) : 300
Maximum pressure (MPa) : 1
Torque capacity of Brake (N.m) : 17027.15
Actuating force (N) : 36.48563
Reaction force (N) : 150
To decrease actuating force increase diameter and friction coefficient

Do You want to continue or Not : N



BAND BRAKE DESIGN

OUTPUT !

Direction of drum :Cw
Contact angle(deg): 270

Reaction force
P1 (N) : 150.00
Actuating force
P2 (N) : 36.49
Torque capacity
T (N.m) :17027.15

Press any key to continue

Example 4: A disc type truck brake has an outer diameter of 400 mm. The material used has a thickness of 3 mm and a lining angle of 40^0. If the speed of truck is 60 km/h, diameter of wheel is 750 mm, duration time per braking is one second and the load acting on each pad is 2000 kg, determine the life of friction material.

BRAKE DESIGN

Options : 1-> Car 2-> Truck 3-> Crane 4-> Hoist 5-> Elevator
Application Field of Brake : Truck
Type : 1-> Front Disk-Rear Drum 2-> Front-Rear Disk 3-> Front-Rear Drum
Type of the Truck Brake : Disk
Velocity of Truck (km/h) : 60
Diameter of Wheel is (mm) : 750

Friction material	Friction coefficient	Temperature	Pressure
1> Synthetic reinous cotton web	0.4-0.6	150	1.17
2> Fenol-Synthetic reinous	0.25	150	0.68
3> Synthetic reinous asbestos web	0.3-0.5	300	1.92
4> Asbestos synthetic reinous with hydraulic pressing	0.2-0.35	500	7.84

Select friction material : Fenol-sr

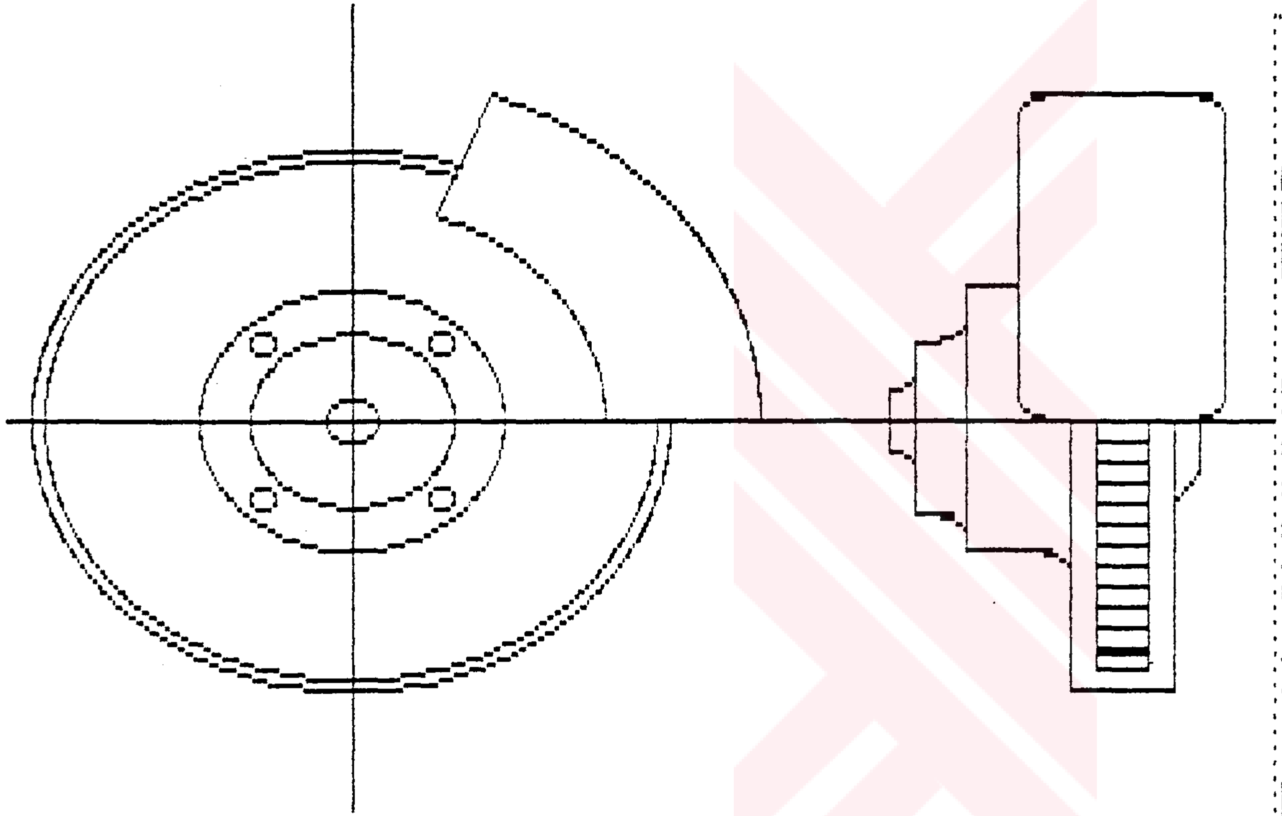
Coefficient of friction : .25
Load acting on each pad (kg) : 2000
The angle of pad lining (deg) : 40
The outer radius of pad (mm) : 400
Do you want to Servo effect or Not : N
The Pad is new or Not : N

Type	b/d (kg/cm^2)	KU ($hp.10^3$)	KT	Commands
1> Clutches, Drum, Band	0.15-0.3	2.0-8.0	1.0-1.6	J=1
2> Disc, J=1 to 2	0.15-0.3	2.0-5.0	1.0-1.6	
3> J=1 to 4	0.1-0.25	0.8-3.5	0.45-0.65	Dry or oil dressed

The b/d ratio : .2
The constant KT : 1.3
Duration time per braking (sec) : 1
Number of braking per hour : 60
Thickness of friction material (mm) : 3
Torque of friction force (kg.cm) : 306180.1
Friction work in each braking (kg.m) : 680.3502
Friction power (hp) : 0.15119
Average diameter of disc (cm) : 51.77915
Life of friction material (hour) : 445686.8

Do You want to Continue or Not : N

DISK BRAKE



INFORMATION

Inner Dia: 200
Outer Dia: 400
Material :
H.Grade Pearlit
 Gray cast Iron
Surface quality
0.000762 mm CLA
P.Angle : 40
Pad material
Fenol-sr
Pad Thick : 3
Capacity :
 306180.1
Life :
 445686.8
Self Energizing
Never Occurs

Press any key to Continue

CHAPTER IX

DISCUSSION AND CONCLUSIONS

9.1 INTRODUCTION

The development of the study are discussed in section 9.2. Conclusion of the work is given in section 9.3. Section 9.4 is devoted to the recommendations for future study.

9.2 DISCUSSION

A Computer Aided Design (CAD) package for the design of machine elements has been prepared and the use of each module is illustrated by various examples in this study. The developed CAD package is designed in a modular form. With this feature, the user is given an opportunity to run each module independent from the others, if required. The work reported in this thesis includes seven types of machine elements, namely; shafts, anti-friction bearings, journal bearings, bolted joints, helical springs, V-belts, and brakes. Although there are other types of machine elements, they could not be included in the package, due to memory limitation, since they require a huge amount of memory for their individual databases. But, one can easily integrate the other machine elements because it provides a modular structure.

Although there are many studies in the literature related on the design of machine elements, only a few are on the CAD. The programs are arranged in such a way that undergraduate mechanical engineering students may use these programs for educational purpose.

Running of the package is extremely simple and little skill is required. The program is written in QuickBASIC programming language. A menu is developed by which for enables the user to interact with the system.

9.3 CONCLUSION

The main purpose of this study was to prepare a whole, compact and user friendly **Computer Aided Design** package for the design and selection of machine elements. This has been achieved by preparing a distinct module (program) for each of the machine elements and driving them under a supervisory program. The functions of a Computer Aided Design package, including the selection as well as the design and analysis of machine elements are fulfilled by this work.

With the computer program developed package, it would naturally be possible to reduce the time of design of machine elements which is one of the main goal of CAD applications. The machined elements which are designed by using the developed package have been checked many times whether they are conservative or not, and positive results have been obtained.

9.4 RECOMMENDATIONS FOR FURTHER STUDY

The work presented in this thesis can be extended to the following fields;

- The developed CAD package has a modular structure. So, it can easily be extended and modified. The experience gained from this study can be applied to other machine elements like gears, screws, welded joints and chains, etc., in order to develop more sophisticated and complete Computer Aided Design package of machine elements.

- Optimisation of those designs could be included.

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APPENDIX 1

The endurance limit of test specimen is expressed as,

$S_e' = 0.5\ S_{ut}$ if $S_{ut} < 1400$ Mpa

$S_e' = 700$ Mpa if $S_{ut} > 1400$ Mpa

The prime work on S_e' refers to the rotating-beam specimen itself. This limit is needed to be modified for the designed machine element. Using this idea, it can be written

$$S_e = k_a k_b k_c k_d k_e k_f S_e' \tag{A.1}$$

where S_e is the endurance limit of mechanical element, k_a is the surface factor, k_b is the size factor, k_c is the reliability factor, k_d is the temperature factor, k_e is the modifying factor for stress concentration, k_f is the miscellaneous-effects factor.

Surface Factor, k_a depend upon the quality of the finish and the tensile strength. For polished surfaces, $k_a = 1$ and the formula derived from examined data for other surfaces is;

$$k_a = a S_{ut}^b \tag{A.2}$$

where a and b are to be found in Table A.1.

Table A.1. Surface finish factors.

Surface Finish	Factor a	Exponent b
Ground	1.58	-0.085
Machined or Cold-drawn	4.51	-0.265
Hot-rolled	57.7	-0.718
As forged	272.	-0.995

Size factor, $k_b = 0.6$ for axial loading and for bending and torsion is found from

$$k_b = 1\,, \qquad \text{if } D \le 8 \text{ mm}$$

$$k_b = 1.189D^{-0.097}, \qquad \text{if } D > 8 \text{ mm} \tag{A.3}$$

If a point is subjected to any load combination in which axial load is present, a correction factor α, which is defined as the ratio of k_b for bending or torsion and k_b for axial loading must be used. In this load combination, k_b for bending or torsion is taken for calculation of the endurance limit of mechanical element and the axial alternating component at this point is multiplied by α. For stationary shafts, equivalent diameter must be calculated first, from the equation given below.

$$D_{eq} = 0.370D \tag{A.4}$$

The reliability factor k_c corresponding to an 8 % standard deviation of the endurance limit is found from Table A.2.

Table A.2. Reliability factor k_c.

Reliability R	Standard variable z_r	Reliability factor k_c
0.50	0	1.000
0.90	1.288	0.897
0.95	1.645	0.868
0.99	2.326	0.814
0.999	3.090	0.753
0.999 9	3.719	0.702
0.999 99	4.265	0.659
0.999 999	4.753	0.620

From the effect of temperature on the endurance limit of a group of carbon and alloy steels, an approximate temperature factor k_d can be selected as

$$k_d = 1 , \quad \text{if } T \leq 350\ °C$$

$$k_d = 0.5 , \quad \text{if } 350\ °C < T \leq 500\ °C \tag{A.5}$$

The modification factor for stress concentration k_e is found from the relation

$$k_e = \frac{1}{K_f} \tag{A.6}$$

where K_f is the fatigue-strength reduction factor and it is defined as the ratio of endurance limit of notch-free specimen and endurance limit of notched specimen.

Notch sensitivity q is defined by the equation;

$$q = \frac{K_f - 1}{K_t - 1} \tag{A.7}$$

where K_t is the theoretical stress-concentration factor considering the geometry of discontinuity and the type of loading. Tables for stress concentration factors are given in reference [1]. If there are two discontinuities at the same point, K_t at that cross-section is the multiplication of K_t of these geometry. Notch sensitivity q is selected from Table A.3 and A.4 due to the notch radius and the type of the loading. After finding of K_t and q, K_f is found to be;

$$K_f = 1 + q(K_t - 1) \tag{A.8}$$

In the case of combined loading, the several stress-concentration factors differ from each other and a unique value of K_f and k_e does not exist. Solution is to use $k_e = 1$ and multiply alternating components of stresses by their respective K_f values. Miscellaneous effect factor K_f is intended to account for the reduction in endurance limit due to uncountable effects and taken as unity for shafts.

Table A.3. Notch-sensitivity q for materials in reversed torsion. For larger notch radii use the values of q corresponding to r = 4 mm.

Notch radius mm	Annealed steel (Bhn < 200) - q	Quenched and Drawn steel (Bhn > 200) - q	Aluminium Alloys q
0.050	0.400	0.600	0.100
0. 100	0.480	0.800	0.220
0.250	0.600	0.860	0.370
0.300	0.670	0.890	0.460
0.500	0.760	0.915	0.570
0.750	0.820	0.950	0.670
1.000	0.860	0.960	0.715
1.250	0.880	0.970	0.760
1.500	0.900	0.980	0.790
2.000	0.930	0.985	0.840
2.500	0.950	0.990	0.860
3.000	0.960	0.995	0.890
4.000	0.990	0.995	0.910

Table A.4. Notch-sensitivity q for materials in reversed bending or reversed axial loads.

Notch radius - mm	Aluminium alloy	Steels S_{ut}=0.4	S_{ut}=0.7	S_{ut}=1.0	S_{ut}=1.4 GPa
0.100	0.200	0.360	0.540	0.670	0.810
0.150	0.250	0.440	0.590	0.710	0.840
0.250	0.300	0.480	0.620	0.740	0.850
0.350	0.380	0.530	0.640	0.760	0.860
0.500	0.410	0.550	0.670	0.790	0.870
0.625	0.450	0.600	0.700	0.810	0.900
0.750	0.490	0.620	0.730	0.830	0.910
0.875	0.520	0.640	0.740	0.840	0.920
1.000	0.540	0.650	0.750	0.850	0.930
1.250	0.590	0.660	0.760	0.860	0.930
1.500	0.630	0.670	0.780	0.870	0.940
2.000	0.680	0.710	0.810	0.890	0.950
2.500	0.730	0.730	0.830	0.900	0.960
4.000	0.830	0.780	0.860	0.930	0.970