OPTIMAL DESIGN OF POWER TRANSMISSION SHAFTS

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ABSTRACT

OPTIMAL DESIGN OF POWER TRANSMISSION SHAFTS

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In this study, a computer program is developed for analysis and design of any type of a shaft, such as hollow or solid, stepped or uniform, circular or noncircular which is subjected to single and combined, steady and alternating bending, torsion and axial loads which can be integrated with other machine elements.

For such a shaft , the problem is fundamentally a fatigue design with constraints, such as shaft speed limitations. In such a case dynamic analyses of the shaft must be carried out.

Furthermore, in this study, in addition to shaft speed limitations due to the critical speeds of lateral and torsional loads dynamic responses due to various alternating loads are considered. These limitations are generated as new design constraints for optimal design purpose.

Keywords:Power Transmission Shafts, Vibration Modeling, Fatigue Design, Dynamic Modeling, Computer Aided Design, Optimum Design

ÖZET

GÜÇ ILETIM MILLERININ OPTIMUM TASARIMI

Atay, A. Necip

Yüksek Lisans,Makina Mühendisliği Bölümü

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Şubat 1998, 129 sayfa

Bu çalışmada tek yönlü, birleşik, durağan, değişken türdeki eğilme burulma ve eksenel yüklere maruz kalan, içi boş veya dolu, düzgün veya adımlı, yuvarlak veya yuvarlak olmayan bütün mil türlerinin tasarlanması için kullanılabilecek bir yazılım geliştirilmiştir.

Mil tasarımında problem genellikle yorulma tasarımı çerçevesinde çözülebilir, ancak, bazı durumlarda tasarımcı mil hız sınırlarını gözönünde bulundurmak zorundadır. Böyle durumlarda yorulmaya göre tasarım yeterli değildir. Aynı zamanda milin dinamik analizinin yapılması gerekir.

Sunulan çalışmada yalnız eğilmeye ve burulmaya göre olan sistemlerin kritik hızları değil, aynı zamanda, değişken yüklerin dinamik tepkileri tasarımda göz önünde bulundurulmuştur. Bu sınırlandırmalar milin optimum tasarımı çerçevesinde tekrar ele alınmıştır.

Anahtar kelimeler : Güç İletim Milleri, Titreşim Modellemesi, Yorulmaya Göre Tasarım, Bilgisayar Destekli Tasarım, Optimum Tasarım.

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Finally, the greatest thanks are to my parents for their never ending support and reliance through my education.

...to my family

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CHAPTER 1

INTRODUCTION

1.1 GENERAL

There is no doubt that computers are invaluable in design and production engineering. Their benefits are well and widely documented. Its speed of doing complex calculations enable a designer to try many possible design solutions. Computer technology has now become a useful tool and created new design procedures. Its capability of doing complex calculations enable a designer to try many possible design solutions and to select the optimal one.

This study is a part of gear-shaft-bearing design project which can be called as power transmission systems. The complete project is composed of some subsystems which is called functional modules. The development of the shaft module which is shown in Figure 1-1 is the main scope of the study.

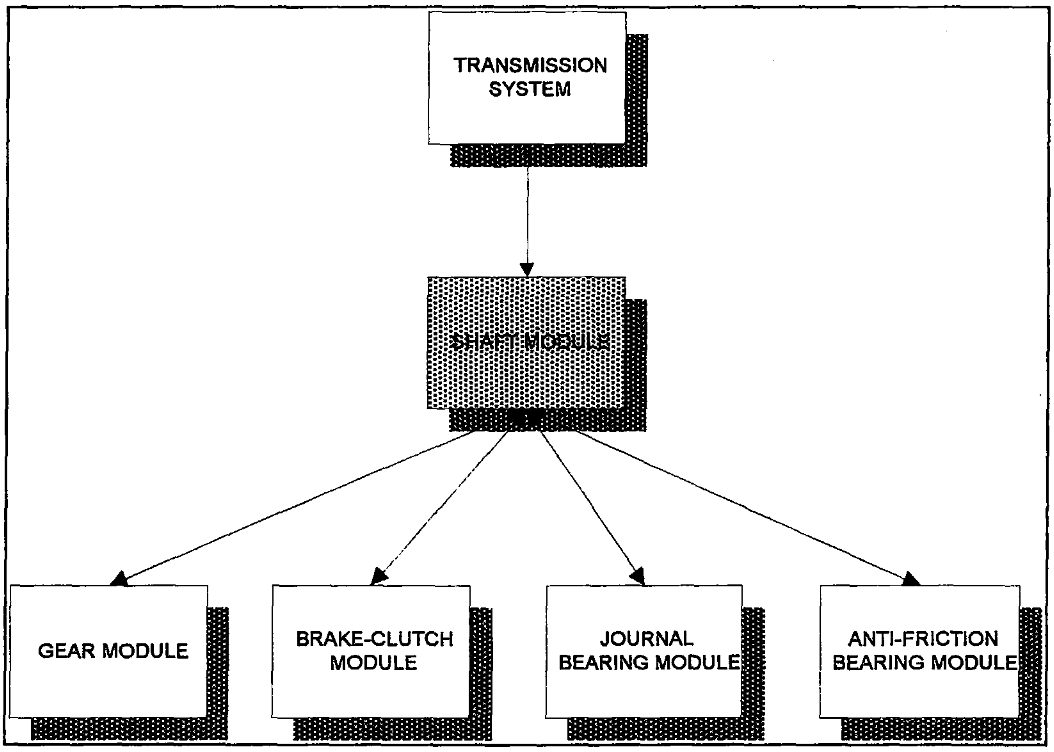


Figure 1-1 Decomposition of the System

The objective is preparing a program that can be used to design any type of shaft such as hollow or solid , stepped or uniform , circular or noncircular, subjected to single or combined steady and alternating bending , torsion and axial loads. The goal in the design is to calculate shaft diameters so that it does not fail under loading and does not deflect above allowable limits. This traditional design procedure consists of first adopting intuitively the geometry and materials of the structure and then calculating for specified design loads the value of behavioral or state variables such as stresses and displacements. After successive modification of geometry, this procedure is then repeated until calculated behavior satisfies certain prescribed requirements which are usually expressed in the form of inequalities representing upper limits on the stresses and the displacements or a lower limit on the load capacity.

In the second part, it is aimed to analyze and predict the stability of the system which is an important part of modern technology. The model and the program is developed so that shafts with any configuration and with any number of machine elements can be analyzed. A mathematical model which utilizes Influence Coefficient Method is developed in order to analyze flexural behavior of the system. For torsional behavior, Holzer Method which is very commonly used and takes less processing time is preferred. Free and forced response analyses of system under alternating loads and unbalances are considered.

The disadvantages of the foregoing procedure is obvious. The design can be highly uneconomical even when an intuitively selected solution satisfies the behavioral constraints. Until recently, however , there has been relatively little work in the field of optimal structural configurations, the economical importance force engineers to do so. With these methods, the designer can evaluate more alternatives, thus resulting in a better and more cost-effective design.

In what is defined in optimal design, first the required structural behavior together with design loads and geometrical constraints are specified and a further quantity termed as objective function is also formulated. The aim of the subsequent computational effort is to select the geometry of the shaft so that the required behavior is achieved at the optimum conditions.

The program is mainly written in the C++ language. In order to satisfy modularity, flexibility and multienvironment support user interface is separated and developed by using Visual Basic.

1.2 SURVEY OF LITERATURE

A number of theoretical shaft design methods have been discussed by many authors. H. Burr [1] discussed the design method in which the shaft is under the simple or combined static and fluctuating stresses. Similar formulas were given by Deutschman [2]. A number of approaches are proposed for the fatigue life of the shaft is presented by Shigley [3] which include Soderberg ,General Biaxial Stress with Modified Goodman , Sines and Kececioglu . These approaches are used with the combination of fatigue failure theories which are distortion energy and maximum shear stress theory. Maday [4] made an investigation on the design of stepped shafts for minimum volume and minimum rotating inertia. The fatigue factors in addition to bending moments , torque and load limit factors have been included in the role of transmission shaft failure by NASA [5].

Walton, Prayoonrat and Taylor [6] described an interactive design program, CAFA in their papers. This program enables transmission drives, motor shafts axles and machine tool spindles to be analyzed in attempt to ensure adequate fatigue life. This paper reviews fatigue analysis particularly for the design of power transmission drives.

As an application of Computer-Aided Design Nguyen [7] developed a computer program in FORTRAN for general convenient design of the power shafts under steady and alternating loads. He has included the capability of designing noncircular shafts into the program. Also designing a shaft for minimum weight subject to design constraints was available.

In Turkey, in the Mechanical Engineering Department of Middle East Technical University, Türkmen and Kaftanoğlu [8] carried out a research on the Computer-Aided Design of gear transmission for parallel shafts. Another application for perpendicular shaft was done later by Ergür and Kaftanoğlu [9]. After that a method and a computer program is developed for design of power transmission shafts and for selection of antifriction bearings by Arıkan [10]. In that study results were used to determine the most suitable solution according to criteria such as minimum cost, minimum power loss or minimum weight. An expert computer program system to design power transmission system is developed by Kaynaköz [11]. In this study previously developed computer programs were integrated to develop an expert computer program system on an IBM-AT compatible personal computer by using FORTRAN 77 language. A similar computer program to design mechanical power transmission systems was prepared by Çarkoğlu [12]. He has included , the integrated design of power shafts, brake and clutches , spur and helical gears , journal bearings , antifriction bearings to the program. In the study, a collection of machine elements connected to the shaft were designed. The computer programs which were used in this study, was taken from five different theses and modified to work on the HP Apollo workstations under Domain Operating Systems as an integrated system.

A design strategy was developed by Kaftanoğlu, Ulugül and Çarkoğlu [13] for the optimal integrated design of a mechanical power transmission system including the constraints of each member and with the objective to satisfy a global optimum for the system.

Another paper by Fagan [14] that reported progress so far in the development of an expert system that assists the designer in selecting the correct combination of ball and roller bearing to support a shaft for a given set of operating conditions. Also the methodology of a power transmission shaft was illustrated in the paper. The methodology was similar with the strategy developed by Kaftanoğlu , Ulugül and Çarkoğlu[13].

A computer program developed by Azarm [15] to design and optimize a hydraulic dredge hub and shaft of a dual wheel excavator was developed. The hub and shaft assembly were optimized for minimum weight. In the study two different assemblies were examined , the keyed connection and the threaded connection.

Parallel to these , there has been many studies for the vibratory behavior of rotating and torsional power transmitting systems. An extensive survey of linear mathematical models used in gear dynamics analysis were given in the paper by Özgüven and Houser [16].

Dynamic analysis of geared shaft systems composed of two gears, two shafts and two inertias representing the drive and the load was studied by Özgüven and Şener [17] using continuous system models. Later a finite element model and a computer program for dynamic analysis of multi geared rotor systems were developed by Yılmaz [18]. In this study geared rotor system to be analyzed consisted of three rotors which were composed of discrete disks, circular shaft segments of distributed mass, multi bearings, and were coupled by two pairs of gears. Another mathematical model which utilizes Transfer Matrix Method has been developed in order to analyze dynamics of multi stage helical geared rotor systems by Okan [19]. A computer program was written and a parametric study was performed in order to demonstrate the effectiveness of the model.

CHAPTER 2

DESIGN OF POWER TRANSMISSION SHAFTS

•

Shafts are solids of revolution intended for transmitting a torque along their axis and for supporting rotating machine components. Since the transmission of torque is associated with the development of forces applied to the shafts, such as forces acting on the teeth of gear, brake and clutch, shaft are usually subjected to transverse forces and bending moments in addition to the torque.

Shafts can be plain shafts or stepped shafts. The diameter of a shaft along its length is determined by the load distribution, bending moments and torque, axial loads and conditions imposed by the manufacturing and assembly processes used.

Shafts may be hollow in design. A hollow shaft with a hole-to-outside diameter ratio of 0.75 is lighter by about 50% than a solid shaft of equal strength and rigidity.

The transition between two shaft steps of different diameters is usually designed with a fillet of a single radius. The radius of the fillet is to be taken as smaller than that of the edge round or radial dimension of the chamfer of the components to be mounted on the shaft step against the shoulder. It is desirable to have the fillet radius of heavily stressed shafts larger or equal to 1/10 of the shaft diameter.

2.1 CONVENTIONS AND ASSUMPTIONS

Shafts are usually designed as beams on hinged supports. This assumption corresponds with sufficient accuracy to the actual conditions. Forces are transmitted to a shaft through the components mounted on it: toothed gears, brakes, clutches, couplings, etc. For simplest calculation it is assumed that the components transmit forces and torques to the shaft at the middle of their width , and the design is based on the corresponding cross-sections.

The first step in analysis of a shaft is dividing the shaft into some stations. Stations are the nodes that shaft should be defined by the user with their components on it. In Figure 2-1 a sample shaft that is defined by these nodes are shown. Distributed loads are usually equally lumped at the nodes. In most cases this is adequate and also it simplifies stress analysis since with concentrated masses, the maxima of shear forces and bending moments exist at the nodes. Therefore analysis between nodes is not necessary for stress analysis. For deflection analysis this is not true but placing some nodes in the vicinity of the places where the maximum deflection is expected, is usually adequate.

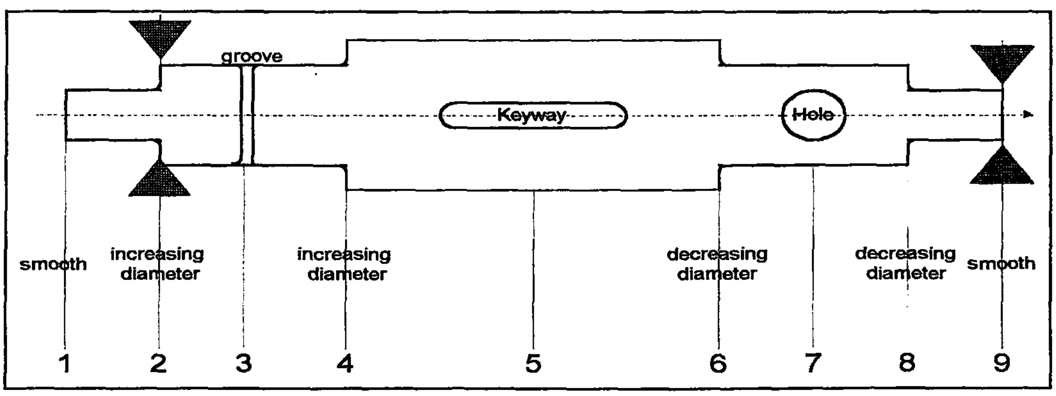


Figure 2-1 A Shaft Sample with Stations

Shafts with continuously variable cross section can be analyzed by way of taking enough nodes and considering constant section properties between nodes. In order to take this effect into consideration, shaft is divided into some substations which is shown in Figure 2-2 where odd numbers symbolize the left side of the original stations and the even numbers symbolize the right. Also this kind of application in the calculations makes it easier in drawing graphs which are not continuous like the moment diagrams.

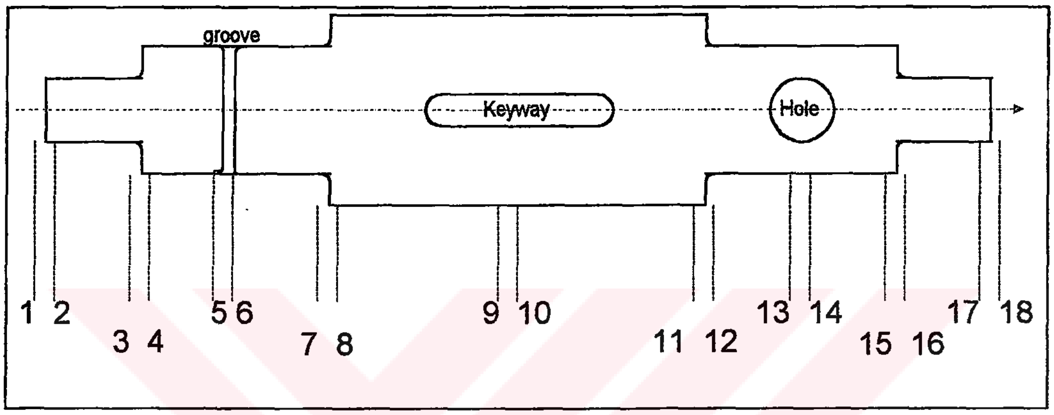


Figure 2-2 Sample Shaft with Its Substations

In order to determine the bearing reactions, loading, shear force, bending moment, axial force and torque diagrams of the shaft, there is a need of deciding a sign convention in axis. Figure 2-3 shows the positive directions of the forces that are used in calculations.

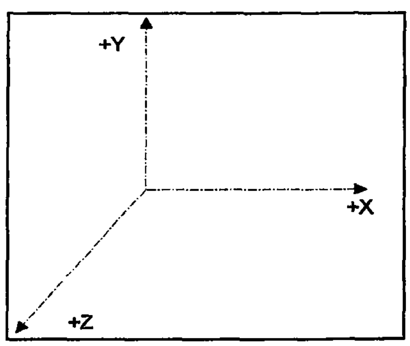


Figure 2-3 Sign Convention of Forces

Other sign conventions that are used for finding the torque and the bending moment distributions around the shaft are shown in Figure 2-4 and Figure 2-5, respectively. Convention for the torque is decided while looking from the left hand side of the shaft.

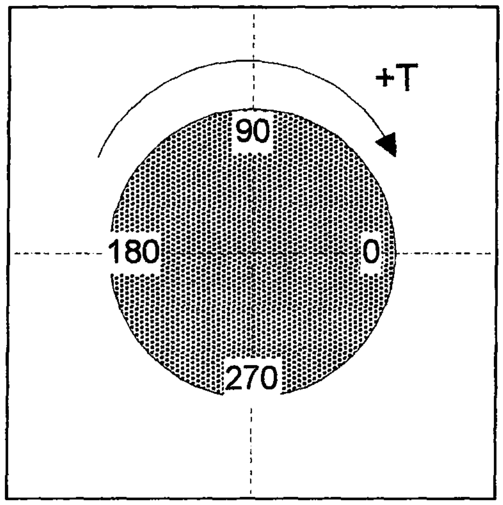


Figure 2-4 Sign Convention of Torque

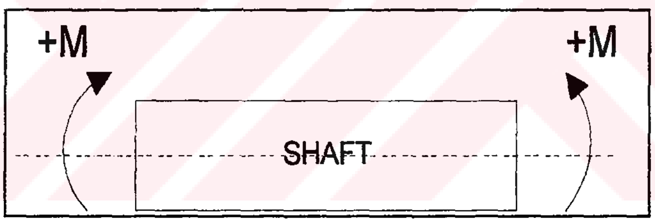


Figure 2-5 Sign Convention of Moment

Vibration analysis of shafts can be conveniently performed with computer methods, in particular by using influence coefficient method for the case of lateral vibration and Holzer method for the case of torsional vibration.

A typical geared rotor system consists of the following elements :

- shafts
- rigid disks
- flexible bearings

In vibrational systems, main difficulty is the solution of the eigenvalue problem. For large systems by using numerical techniques it is very difficult to find all of the eigenvalues and it takes time even with a computer. This is why some simplifications without forgetting our main purpose are needed. Since our main purpose to prepare a shaft design program that can be integrated with other machine design projects, processing time is very important. Another reason and the best one is the memory problem which occur while opening arrays with large dimension.

In order to avoid non-symmetric system matrices which result in complex eigenvalues, gyroscopic effect is ignored. Damping coefficients of bearings and shaft are not included as these coefficients double the size of matrices. Since the gear mesh causes coupling between the torsional and transverse vibrations of the system , it is not very suitable for our aim.

In the above discussion, both for lateral and torsional vibration of shafts, the masses and inertias are considered concentrated at the nodes. For most practical purposes this procedure is accurate enough and simple. The mass and inertia of shaft is equally divided at the nodes where discs are added to these (if there is any). In the case of flexible supports a linearized model with one spring coefficient for each bearings are used.

2.2 DESIGN CRITERIA

Adequate rigidity and strength are the main criteria for shaft design. Rigidity is needed to maintain good contact in gears, to have uniform oil films in gears and journal bearings, and to avoid shortening of life in rolling contact bearing. Strength is needed so that shaft will not fracture or yield. Stability is another design criteria as loads will be amplified 3 or 20 times their nominal value while the shaft speed passes through a lateral or torsional natural frequency. Even running 20 percent away from a natural frequency can double nominal loads.

2.3 DESIGN OF SHAFTS FOR STRENGTH

Figure 2-6 illustrates many of the terms used in fatigue analyses and shows the typical performance of steel under frequently applied loads. The stress , S to couse failure is plotted against the number of cycles, N , of stress reversal to failure, to give the S-N curves as shown in the figure. As the bending moment is reduced, the number of cycles to couse failure increases. For steels, if the specimen survives more than about 10^6 cycles, the part is unlikely to fail even if the test continues indefinitely. The stress corresponding to the point at which curve becomes parallel to the abcissa is called endurance limit, S_e. Similar curves can be obtained for a specimen subjected to reversed torsion, where the corresponding endurance limit in shear is denoted as S_{es}.

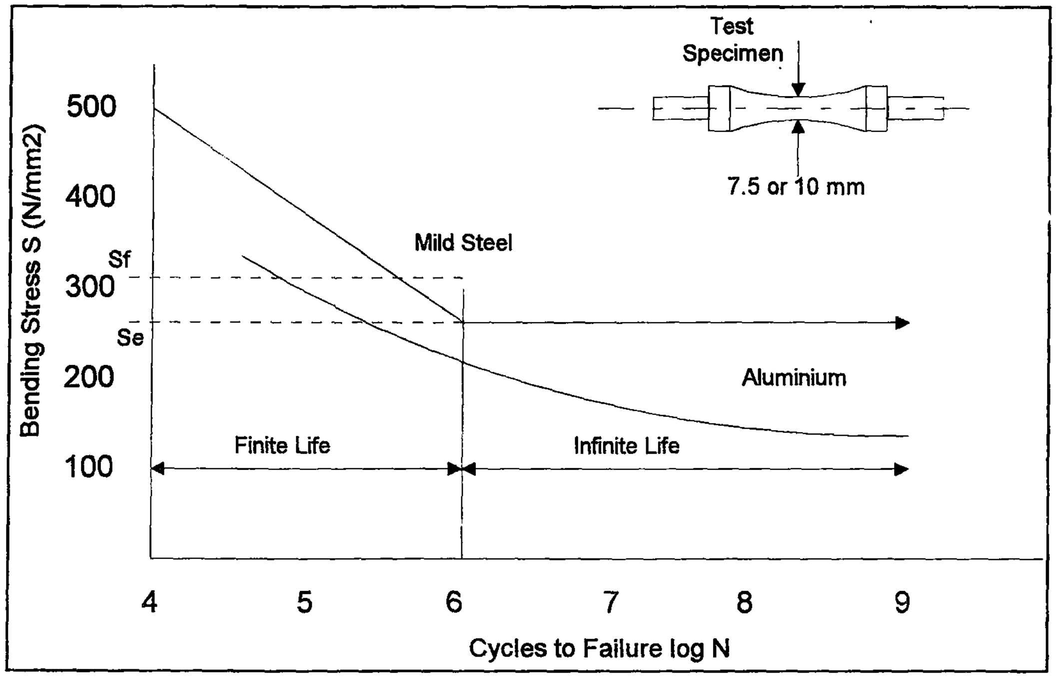


Figure 2-6 S-N Diagrams from Endurance Tests

The important point as far as the design of a component is concerned , is that the endurance limit is well below the ultimate and yield strength of the material S_u and S_y . Thus machined part designed on the basis of static stress analyses and with an apparently adequate factor of safety can fail under relatively low stresses.

2.3.1 FATIGUE STRENGTH CALCULATIONS

The main loads acting on shafts are forces due to power transmission. Loads constant in magnitude and direction cause constant stresses that vary in an alternating symmetrical cycle in rotating shafts. On the other hand loads alternating in magnitude are added to the mean components for conservative design. Constant loads rotating together with the shafts , due to unbalanced rotating components , cause constant stresses.

Any rotating shaft subjected to steady and alternating loads end up with alternating and mean components of normal stresses and shear stresses.

The distortion energy theory is concerned mainly with predicting the onset of yielding. According to this criterion, yielding will occur when the strain energy of distortion / unit volume equals the strain energy of distortion / unit for a specimen in uniaxial tension or compression. For a biaxial stress state this criterion can be expressed by :

$$S_y^{\,2} = \sigma_1^{\,2} - \sigma_1\sigma_2 + \sigma_2^{\,2} \qquad 2\text{-}1$$

In two dimensional case, there are normal stresses σ_x and σ_y and the shear stress τ_{xy} have both alternating and mean components. In this case two Mohr's circle may be constructed for the alternating and mean components. Then the corresponding principal stresses for alternating and mean components are obtained. By using Von-Misses stresses following equations are obtained.

$$\sigma'_a = \sqrt{\sigma^2_{xa} - \sigma_{xa}\,\sigma_{ya} + 3\tau_{xya}^{\,2}} \qquad 2\text{-}2$$

$$\sigma'_m = \sqrt{\sigma^2_{xm} - \sigma_{xm}\,\sigma_{am} + 3\tau_{xym}^{\,2}} \qquad 2\text{-}3$$

In the case of rotating shafts , these equations reduce to:

$$\sigma'_a = \sqrt{\sigma_a^{\,2} + 3\tau_a^{\,2}} \qquad 2\text{-}4$$

$$\sigma'_m = \sqrt{\sigma_m^{\,2} + 3\tau_m^{\,2}} \qquad 2\text{-}5$$

The formulations against this general form of loads are given in the Figure 2-7 where :

M_a: alternating bending due to radial static + alternating loads for conservative design

M_m: mean or static bending due to unbalances

T_a : alternating torsion

T_m : mean or steady torsion

P_a : alternating axial load

P_m : mean or static axial load

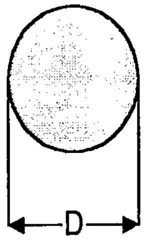
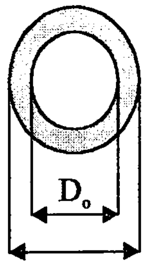
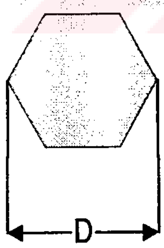
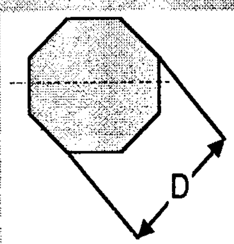
CROSS SECTION	MAXIMUM NORMAL STRESS	MAXIMUM SHEAR STRESS
	$\sigma_a = \frac{32K_fM_a}{\pi D^3} + \frac{4K_{fa}P_a}{\pi D^2}$ $\sigma_m = \frac{32M_m}{\pi D^3} + \frac{4P_m}{\pi D^2}$	$\tau_a = \frac{16K_{fs}T_a}{\pi D^3}$ $\tau_m = \frac{16T_m}{\pi D^3}$
	$\sigma_a = \frac{32K_fM_aD_o}{\pi(D_o^4 - D_i^4)} + \frac{4K_{fa}P_a}{\pi(D_o^2 - D_i^2)}$ $\sigma_m = \frac{32M_mD_o}{\pi(D_o^4 - D_i^4)} + \frac{4P_m}{\pi(D_o^2 - D_i^2)}$	$\tau_a = \frac{16K_{fs}T_aD_o}{\pi(D_o^4 - D_i^4)}$ $\tau_m = \frac{16T_mD_o}{\pi(D_o^4 - D_i^4)}$
	$\sigma_a = \frac{6K_fM_a}{\pi D^3} + \frac{K_{fa}P_a}{\pi D^2}$ $\sigma_m = \frac{6M_m}{\pi D^3} + \frac{P_m}{\pi D^2}$	$\tau_a = \frac{4.8K_{fs}T_a}{\pi D^3}$ $\tau_m = \frac{4.8T_m}{\pi D^3}$
	$\sigma_a = \frac{12.8K_fM_a}{\pi D^3} + \frac{8K_{fa}P_a}{3\sqrt{3}\pi D^2}$ $\sigma_m = \frac{12.8M_m}{\pi D^3} + \frac{8P_m}{3\sqrt{3}\pi D^2}$	$\tau_a = \frac{8.75K_{fs}T_a}{\pi D^3}$ $\tau_m = \frac{8.75T_m}{\pi D^3}$
	$\sigma_a = \frac{11.55K_fM_a}{\pi D^3} + \frac{\sqrt{2}K_{fa}P_a}{\pi D^2}$ $\sigma_m = \frac{11.55M_m}{\pi D^3} + \frac{\sqrt{2}P_m}{\pi D^2}$	$\tau_a = \frac{6.342K_{fs}T_a}{\pi D^3}$ $\tau_m = \frac{6.342T_m}{\pi D^3}$

Figure 2-7 Alternating and Mean Stresses Due to Loads

For a part in pure shear , the distortion energy theory predicts that an increase in strength of 15% compared to that predicted using the maximum shear stress theory. The distortion energy theory is in good agreement with data from tests carried out on parts subjected to combined loads, and gives a competitive design solution. It is however, relatively complex to use.

There are many design procedures in literature. Figure 2-8 shows the fatigue design theories for components subjected to varying stress and mean stress. Deriving the equations for factor of safety is an appropriate way of finding the safety levels at the stations. By the way, the strength distribution of shaft can be calculated in terms of factor of safety.

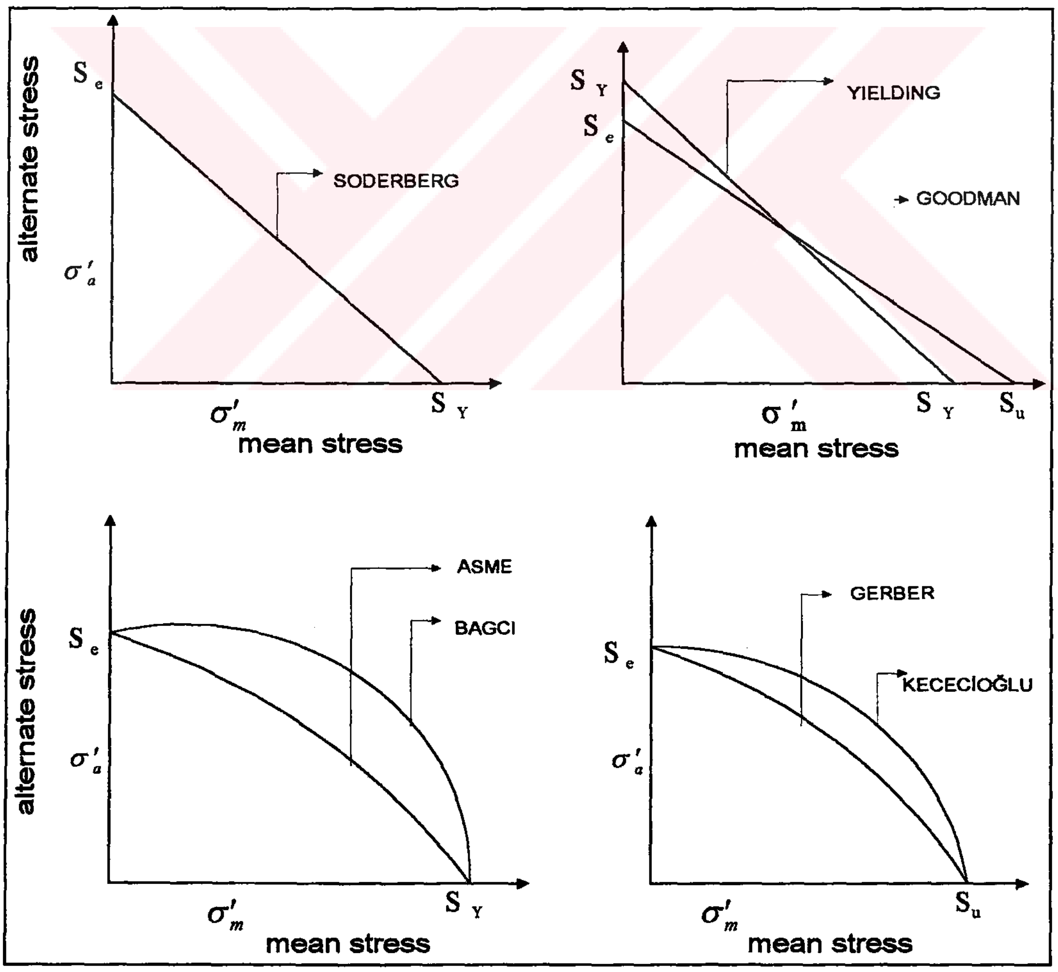


Figure 2-8 Shaft Design Theories

Due to their relative simplicity Soderberg, and Goodman are used frequently in engineering design. On the other hand Langer (Yielding) approach is commonly used for checking the other theories. As they are all straight lines deriving them for the factor of safety is very easy and straightforward.

Although other theories like Gerber , Keçecioğlu and ASME fits the data points more accurately , the fundamental equations are nonlinear and this makes them difficult to use. At that point a numerical method is suitable to use.

General case of the formulation given in literature due to function of safety is in the form:

$$f(N) = N^m A^m + N^n B^n - 1 \qquad \text{2-6}$$

If this formulation is generated by using Newton-Raphson method :

$$N_{new} = N_{old} - \frac{f(N_{old})}{f'(N_{old})} \qquad \text{2-7}$$

$$N_{new} = N_{old} - \frac{N_{old}{}^{m} A^m + N_{old}{}^{n} A^n - 1}{mN_{old}{}^{m-1} A^{m-1} + nN_{old}{}^{n-1} A^{n-1}} \qquad \text{2-8}$$

where A, B, m ,n for the various shaft design approaches are given in the Table 2-1

Table 2-1 Guidelines for Using Fatigue Theories

THEORY	A	B	a	b	NEED CHECK
SODERBERG	$\frac{\sigma_a'}{S^*_e}$	$\frac{\sigma_m'}{S_y^*}$	1	1	NO
GOODMAN	$\frac{\sigma_a'}{S^*_e}$	$\frac{\sigma_m'}{S_u}$	1	1	YES
LANGER	$\frac{\sigma_a'}{S_y^*}$	$\frac{\sigma_m'}{S_y^*}$	1	1	NO
KECECIOĞLU	$\frac{\sigma_a'}{S^*_e}$	$\frac{\sigma_m'}{S_u}$	2.65	2	YES
GERBER	$\frac{\sigma_a'}{S^*_e}$	$\frac{\sigma_m'}{S_u}$	1	2	YES
BAGCI	$\frac{\sigma_a'}{S^*_e}$	$\frac{\sigma_m'}{S_y^*}$	1	4	NO
ELLIPTIC	$\frac{\sigma_a'}{S^*_e}$	$\frac{\sigma_m'}{S_y^*}$	2	2	YES

Using the relations between tensile yield and shear yield strength for distortion energy theory :

$$S^*_Y = \frac{1}{\sqrt{3}} Sy \qquad \text{2-9}$$

$$S^*_e = \frac{1}{\sqrt{3}} Se \qquad \text{2-10}$$

Some of the theories that are presented in Table 2-1 should be checked and modified since they sometimes exceed the yield as shown in the Figure 2-9.

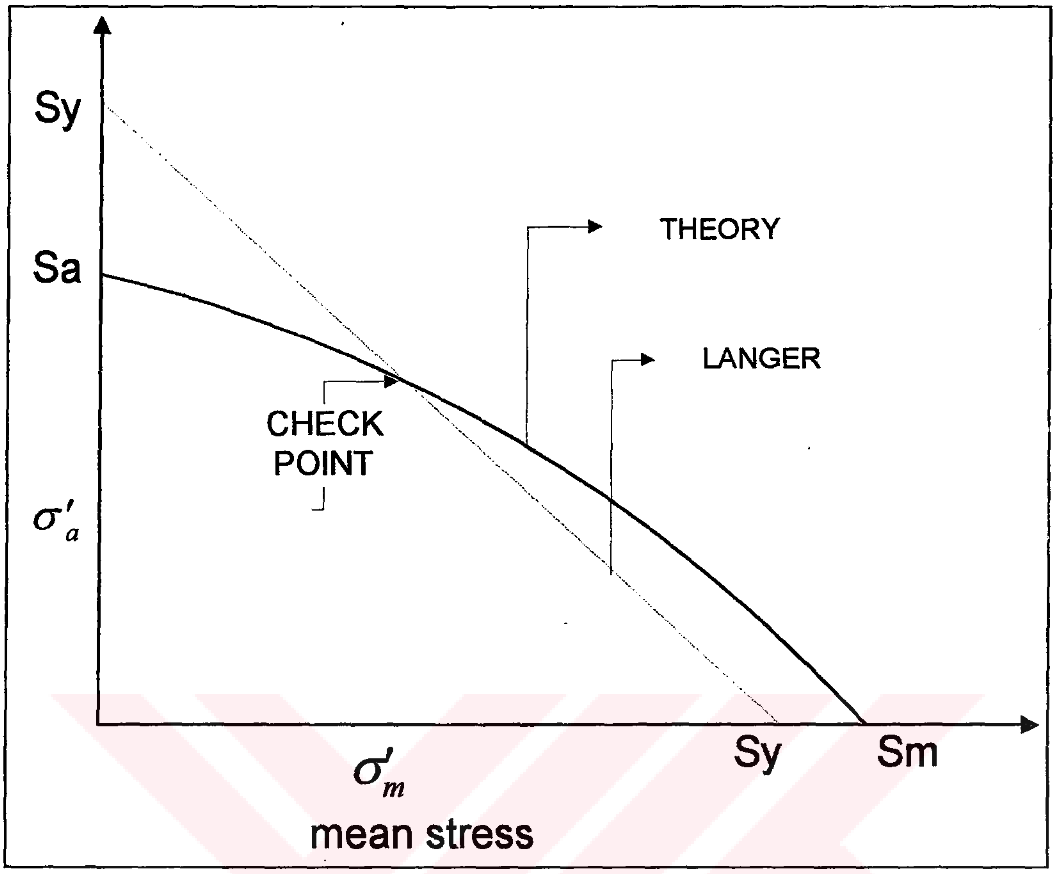


Figure 2-9 General Form of Fatigue Theories

Check is done based on the fundamental equation of Langer Theory. If safety factors that are found by using any of the theories does not specify the condition :

$$\frac{N}{S_y^*}(\sigma'_a + \sigma'_m) \geq 1 \qquad 2\text{-}11$$

then equation for Langer should be used. This kind of check is mentioned in literature as the modified form of the theories.

2.3.2 METHODOLOGY IN FINDING STRENGTH

This section reviews the implementation of the formulations that are presented in the previous section for finding the diameter of shaft that satisfies strength and reviews the methodology step by step.

- After finding alternating and mean components of bending moments, axial forces and torque at each station, calculate the initial diameter by assuming the loads as if they are static and set a step size.

- Find alternating and mean components of shear and normal stresses which are σ_m , σ_a , τ_m , τ_a by using Figure 2-7 for each station.

- Use Von-Mises Theory to find the limiting values of alternating stress σ'_a and mean stress σ_m' by using Equation 2-4 and Equation 2-5.

- Find the safety factors for each station by using the Equation 2-6 if it can be directly found, else use a root finding method (Equation 2-8).

- Check if they specify the condition in Equation 2-11. If it does not specify, then find the factor of safety at that section by using Langer Theory.

- Select the smallest factor of safety which mentions the critical section of the shaft according to the strength and compare it with the users entry. If it is larger increase diameter by adding step size else decrease diameter by subtracting and set the step size to its half.

The flowchart of this methodology is shown in Figure 2-10.

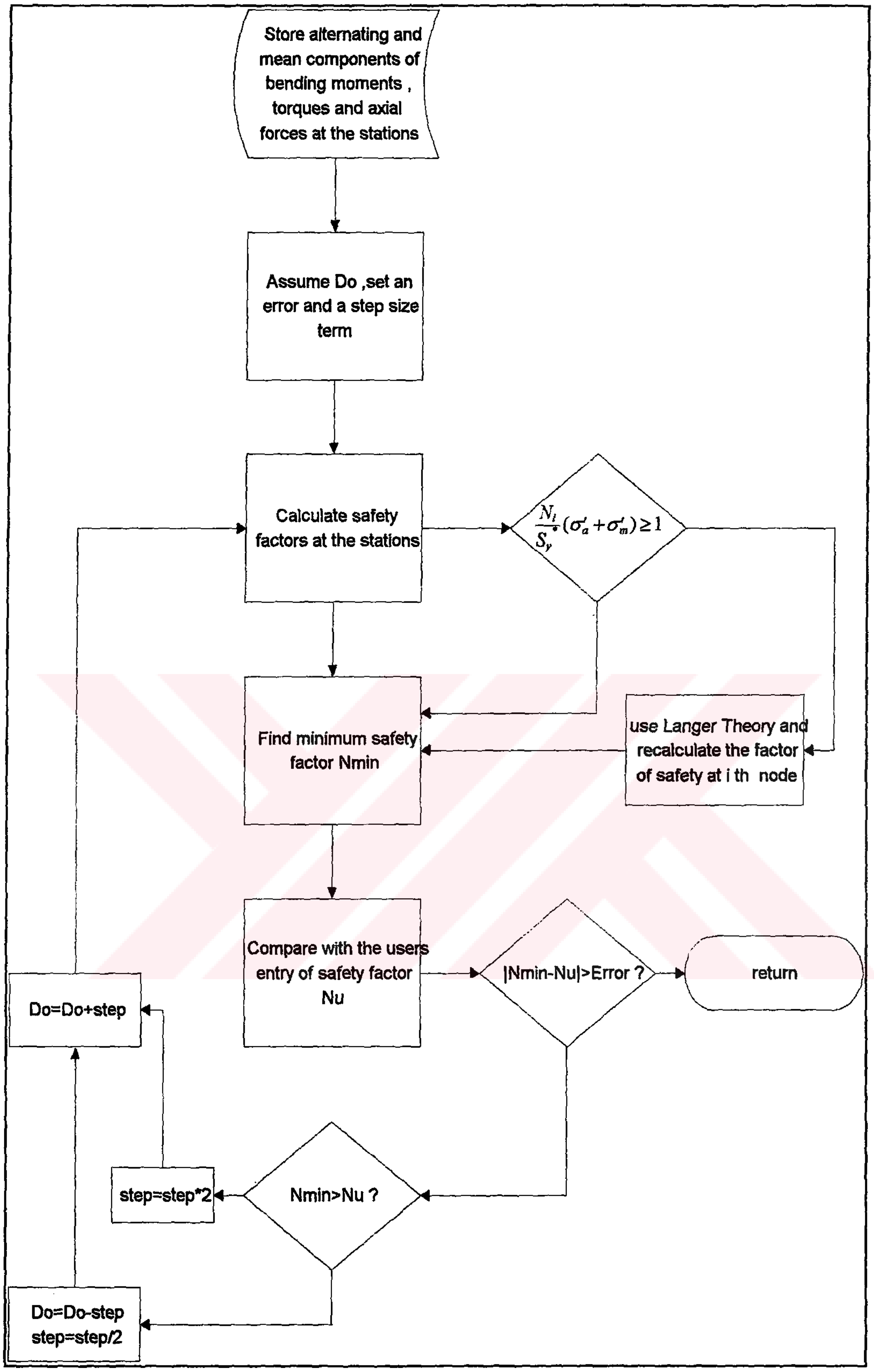


Figure 2-10 Flowchart of Strength Calculations

2.4 DESIGN OF SHAFTS FOR RIGIDITY

The required rigidity of shafts subject to bending is determined mainly by conditions of proper operation of the drive and bearings.

The deflection of shafts has a small effect on brakes and clutches. Therefore they are not usually checked for rigidity. The elastic displacement of shafts carrying toothed gears, leads to angular displacement of meshing gears with respect to each other and consequently load concentration along the face width of the meshing teeth. It also leads to separation of the shaft axes which is unfavorable for gearing. In the case of involute gearing, it shortens the line of contact between the meshing teeth.

The rigidity of shafts running in non-self-aligning sleeve bearings should be sufficient to ensure the required uniform pressure distrubution along the length of the bearing.

The rigidity of shafts running in ball bearing must be such that the balls do not jam due to misalignment of the rings if self-aligning bearings are not used.

Empirical rules are available for checking the allowable deflections and angles of inclination of the elastic lines of shafts. Thus, in speed gearboxes, the maximum deflection of shafts carrying toothed gears should not exceed 0.0002 to 0.0003 of the distance between the supports; the angle of mutual inclination of shafts with meshing gears should be less then 0.001 radian. The angle of inclination in radial ball bearing can be 0.5^o (under the condition that the bearing life is reduced by only 20% and if the bearing unit is manufactured with ordinary accuracy). The same values for roller bearings are 0.2^o and 0.15^o. The maximum deflection of the shafts of induction motors should not exceed 0.1 of the air gap (Dimarogonas [23]).

Also torsional rigidity can be important for various shafts. Static elastic angular deformation of kinematic trains can affect the accuracy of machine performance. The elastic deformations of the drives for low-speed machinery may cause stick-slip motion. For this reason, for example , the angles of twist of long feed shafts of heavy machine tools are limited to values of the order of 0.5^o per meter length. The elastic deformation of divided drives powered from a single motor and used for traversing overhead cranes, portals, cross-members of heavy machine tools, etc. may result in jamming of the slideways. In the transmission shafts of the mechanisms traveling cranes, angles of twist range from 0.15^o to 0.20^o per meter of length. For most shafts, torsional rigidity is of no essential importance and there is no necessity to check the rigidity of such shafts. An allowable angle of twist to 0.25^o per meter length, frequently given in the literature, has become obsolete and cannot be technically substantiated

These relationships are however, are not very specific and cannot be generalized. Rigidity guidelines that are used as default values are presented in Table 2-2.

Table 2-2 Rigidity Guidelines

Case	Maximum Lateral		Maximum Torsional Deflection
	Slope	Deflection	
SHAFTS			
Transmission Shaft	....	0.01 L	
Line shaft	....	0.01 L	
Machinery	....	0.02 L	0.1L or 0.5 L/d
Machine Tool	....	....	
GEARS			
At mesh	0.0005 mm/mm	0.003/0.005	
In general	0.127 mm/(face width)		

2.4.1 DEFLECTION OF SHAFTS

Having determined the shaft diameter required to withstand the severest load conditions, the next step is to calculate the shaft deflection. As it is mentioned before, deflection of shaft is important as its strength. For instances, it is used to establish the minimum permissible clearance between the pulleys, gears and housing for the shaft assembly. The deflection at the gear locations will increase the backlash between the gear teeth, increase the pressure angle and reduce the length of tooth contact.

2.4.1.1 LATERAL DEFLECTION

Lateral deflections and angles of inclination of the elastic lines of shafts are determined by the conventional methods of the strength of materials. For simple cases of design it is expedient to employ design formulas, treating the shaft as a beam (Figure 2-11)

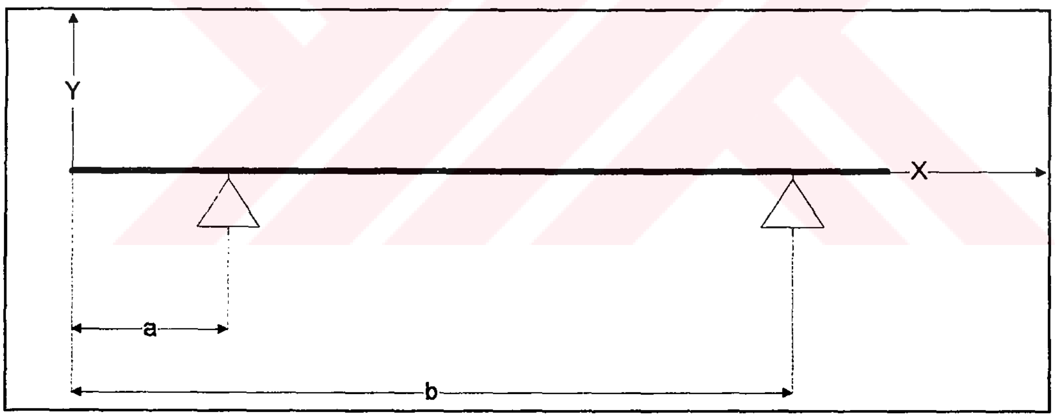


Figure 2-11 Shaft Treated as Beam

The amount of permissible lateral deflection for proper bearing operation, accurate machine tool performance, satisfactory gear tooth action , shaft alignment and other similar requirements may be determined by two successive integration of :

$$\frac{d^2y}{dx^2} = M / EI \qquad 2\text{-}12$$

M = Bending moment

E= Modulus of elasticity

I = Area moment of inertia

The first integration of both sides of Equation 2-13 along the neutral axis, x, yields:

$$\frac{dy}{dx} = \int_0^x \frac{M}{EI} dx + C_1 \qquad 2\text{-}13$$

A second integration over the same interval gives:

$$y = \int_0^x \left(\int_0^x \frac{M}{EI} dx\right) dx + C_1 x + C_2 \qquad 2\text{-}14$$

There are many methods to find the deflection of the power shaft. Besides graphical method, numerical method is the best way to use in interactive computer program.

Since M/EI is a piecewice linear function of x if there are only concentration forces acting on the shaft. Therefore, exact solution may be obtained by using trapezoidal rule.

$$\phi_i = \int_0^x \frac{M}{EI} dx = \frac{(\frac{M}{EI})i + (\frac{M}{EI})i-1}{2}(x_i - x_{i-1}) + \phi_{i-1} \qquad 2\text{-}15$$

Then the slope can be found:

$$\theta_i = (\frac{dy}{dx})_i = \phi_i + C_1 \qquad \text{2-16}$$

The slope is a quadratic function and the use of the trapozoidal rule for a second integration to get the deflection is not exact. Therefore, Simpsons 1/3 rule is used to get a better result.

$$\psi_i = \frac{1}{6}\left[\left(\frac{M}{EI}\right)_i + \left(\frac{M}{EI}\right)_{i-1} + 4\left(\frac{M}{EI}\right)_{i-1/2}\right](x_i - x_{i-1}) + \psi_{i-1} \qquad \text{2-17}$$

Then the deflection can be written :

$$y_i = \psi_i + C_1 x_i + C_2 \qquad \text{2-18}$$

where θ_i and y_i are the slope and the deflection at the i th station.

Locating supports x=a and x=b and specifying "zero" deflection at these supports provides the two conditions for finding C_1 and C_2. The results are :

$$C_1 = \frac{\psi_b - \psi_a}{a-b} \qquad \text{2-19}$$

$$C_2 = \frac{b\psi_b - a\psi_a}{a-b} \qquad \text{2-20}$$

For the shaft subjected to the radial loads and unbalances which are not in the same plane, calculations are done for each plane separately and then they are combined for finding the maximum deflection.

2.4.1.2 TORSIONAL DEFLECTION

The amount of twist permissible depends on the particular application, and varies about 0.08 deg and 1 deg per foot (305 mm). The following equation applies to torsional deflection :

$$\frac{d\,\theta}{dx} = \int_0^x \frac{T}{GJ} dx + C_1 \qquad \text{2-21}$$

where

G = Shear modulus of elasticity

θ = Angle of twist

C_1 = Integration constant refer to the reference plane

If the reference plane is put to the starting point of shaft at the left handside, C_1 =0 can be taken and by using trapezoidal rule to find the exact solution :

$$\theta_i = \left[\frac{(\frac{T}{GJ})_i + (\frac{T}{GJ})_{i-1}}{2} \right] (x_i - x_{i-1}) + \theta_{i-1} \qquad \text{2-22}$$

For portion weakened by keyways, factor of reduction in rigidity, k, is introduced into the right-hand member of the equation :

$$k = \left[1 - (4nh / D\right]^{-1} \;\; ; \;\; \theta_i' = k\theta_i \qquad \text{2-23}$$

where h is the depth of the keyway, n= 0.5 and D is the shaft diameter at the section.

CHAPTER 3

VIBRATION OF POWER TRANSMISSION SHAFTS

The vibratory behavior of rotating and torsional power transmitting systems plays an important role in the design of engineering systems, since detailed predictions of dynamic loads are necessary especially for high speed machinery. Lack of shaft stiffness will result in both linear and torsional vibration, the effect of which can show up in many ways. Not only will the machine perform poorly but this poor performance will effect the quality of products produced by the machine.

Of main practical importance in shaft design is the determination of the natural frequency of vibration to avoid resonance of vibration Observed in shafts are transverse or bending vibrations, angular or torsional vibrations and also combined bending and torsional vibrations. In this study transverse and torsional vibrations are taken into consideration.

The most widely employed are calculations of the fundamental frequencies of vibrations, since these vibrations are usually dangerous because most machines operate under the lowest critical speed. Heavy or high speed machinery, such as turbomachinery, might operate under the second or third.

$$[M]\{q''(t)\}+[C]\{q'(t)\}+[K]\{q(t)\}=\{F(t)\} \qquad 3\text{-}1$$

where $[M]$ is the time-invariant mass matrix and $\{q(t)\}$ is the displacement vector. $[C]$ and $[K]$ are linear time invariant damping and stiffness matrices, respectively. Forcing vector $\{F(t)\}$ represents external excitations.

In the case of our simplification where there is no damping, the equation of motion turns into :

$$[M]\{q''(t)\}+[K]\{q(t)\}=\{F(t)\} \qquad 3\text{-}2$$

3.2 FREE VIBRATION OF THE SYSTEM

The solution of a system of coupled differential equations can be written as the sum of a homogeneous solution and a particular solution. The homogeneous solution reflects the free vibration properties of the system.

In the absence of external forces , $\{F(t)\}=\{0\}$, equation of motion reduces to :

$$[M]\{q''(t)\}+[K]\{q(t)\}=\{0\} \qquad 3\text{-}3$$

which represents a set of n simultaneous homogeneous differential equations. We are interested in a special type of solution of the set, in which all the coordinates have the same time dependence, and general configuration of the motion does not change, except for the amplitude so that the ratio between any two coordinates remains constant during the motion. Mathematically this type of motion is expressed by :

$$\{q\} = \{u\} \ e^{j\omega t} \qquad 3\text{-}4$$

If it is used in Equation 3-3 , equation becomes :

$$[M]^{-1}[K]\{u\} = \omega^2\{u\} \qquad 3\text{-}5$$

which is an eigenvalue problem.

3.2.1 FREE VIBRATION OF THE TRANSVERSE SYSTEM

If the flexibility matrix is available, Equation 3-5 can be converted into a form :

$$[A][M]\{u\}_r = \lambda_r\{u\}_r \quad , \quad \lambda_r = \frac{1}{\omega^2} \qquad 3\text{-}6$$

where $[A]$ is the flexibility matrix which is the inverse of stiffness matrix and λ_r is the real eigenvalue .

These two equations are both available to solve free vibrational behavior of the system. Development of stiffness matrix using influence coefficients is straightforward. However, the calculation of a column of stiffness coefficients for a structural system modeled with n degrees of freedom requires the solution of n simultaneous equations. This leads to significant computation time for systems with many degrees of freedom. Flexibility influence coefficients provide a convenient alternative. They are easier to calculate for structural systems and their knowledge is sufficient for solution of the free vibration problem.

The computation of the stiffness influence coefficients requires the application of the principles of static and some algebraic manipulation. In fact, the generation of n stiffness influence coefficients k_{1j}, k_{2j} ... k_{nj} for any specific j requires the solution of n simultaneous linear equations. Thus n sets of linear equations (n equations in each set) are to be solved to generate all the stiffness influence coefficients of an n degree of freedom system. This implies a significant computational effort for large values of n. The generation of the flexibility influence coefficients, on the other hand, proves to be simpler and more convenient.

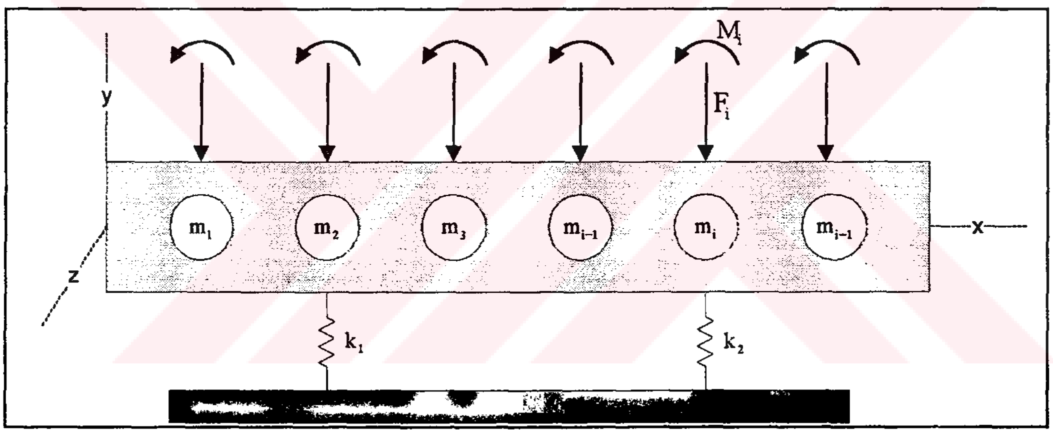


Figure 3-1 Discritized Multidegree of Shaft System

Consider the system shown in Figure 3-1. Let the system be acted one force F_j, and one moment M_j. Let the slope and deflection at point i, (i.e. mass m_i) due to loading be θ_{yi} and y_i. The flexibility influence coefficient, denoted by a_{ij}, is defined as :

$$y_i = a_{ij} \cdot F_j \qquad (3\text{-}7)$$

$$\theta_{y_i} = a_{j(j+n)} \cdot F_j \qquad (3\text{-}8)$$

$$y_i = a_{(i+n)j} \cdot M_j \qquad (3\text{-}9)$$

$$\theta_{y_i} = a_{(i+n)(j+n)} \cdot M_j \qquad (3\text{-}10)$$

In the case of unit forces and moments (j=1,2,....n) act at different points on the system, flexibility matrix can be expressed in form :

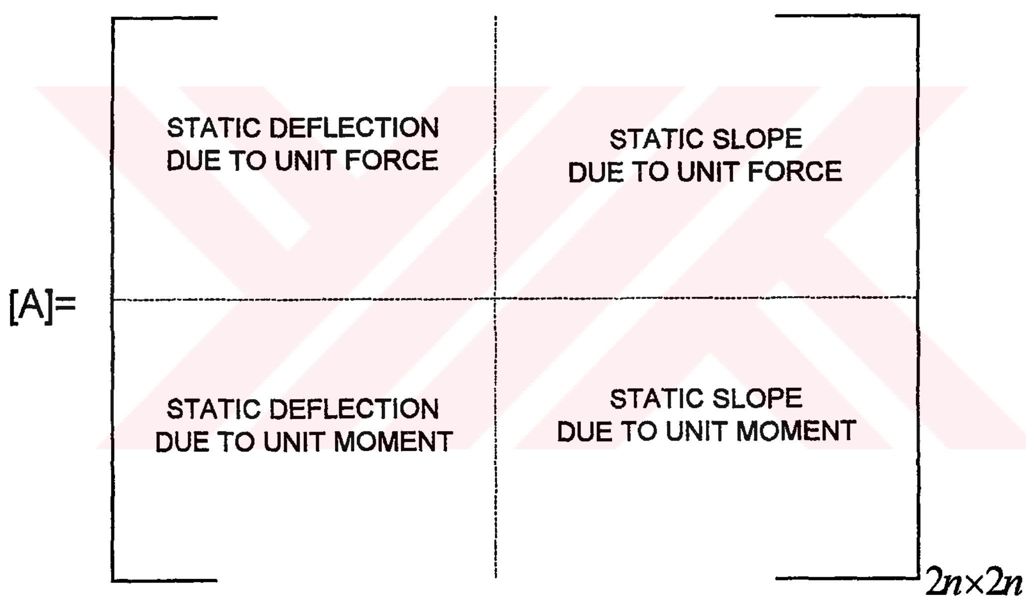


Figure 3-2 Flexibility Matrix

The flexibility influence coefficients of a multidegree of freedom system can be determined as follows.

Considering the flexibility matrix that is mentioned above, static deflection due to unit forces at various stations has to be found. Equation 2-18 can be used to calculate lateral deflection of shafts. This calculation concerns deflections at the bearings are "zero". This assumption does not change the fatigue design purposes very much. Thus, ignoring bearing compliance do not give accurate results especially in finding fundamental natural frequencies of the system. So there is a need of modification in static deflection calculation for the system with flexible bearings.

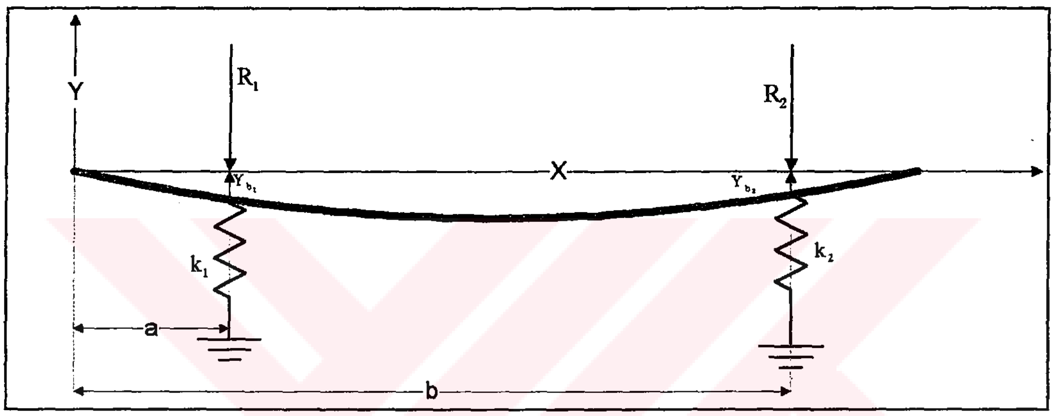


Figure 3-3 Shaft with Flexible Bearings

In Figure 3-3 R_1 , R_2 are the reaction forces acting from the shaft to the bearings. So the following equations can be written for the deflections at the bearings:

$$Y_{b_1} = R_1 k_1 \quad 3\text{-}11$$

$$Y_{b_2} = R_2 k_2 \quad 3\text{-}12$$

k_1, k_2 : Stiffness constants at the bearings

Y_{b_1} and Y_{b_2} : Deflections at the bearings

If the integration constants for the compliance at the bearings are modified, then

$$C_1 = \frac{\psi_b - \psi_a + Y_{b_1} - Y_{b_2}}{a - b} \qquad 3\text{-}13$$

$$C_2 = \frac{b\psi_b - a\psi_a + bY_{b_1} - aY_{b_2}}{a - b} \qquad 3\text{-}14$$

3.2.1.1 SOLUTION OF THE EIGENVALUE PROBLEM

There are various iteration schemes for the solution of the eigenvalue problem, such as the Jacobi method , QR method and the method based on Strum' s theorem. The Jacobi method yields all the eigenvalue and eigenvectors simultaneously. On the other hand , the QR method and the method based on Strum's theorem yield only the eigenvalues, so the eigenvectors must be computed separately. A more efficient technique for computing the eigenvectors and corresponding eigenvalues is inverse iteration. In this study the power method using matrix deflation is decided to solve our eigenvalue problem.

The power method is based on the expansion theorem. The implication of the theorem is that the solution of the eigenvalue problem consists of n linearly independent eigenvectors of $[D]$, which is a dynamical matrix given by Equation 3-5. The expansion theorem implies further that these eigenvectors span the n-dimensional vector space {u} , where {u} represents a possible motion of the system. Hence , any such vector {u} can be expressed as a linear combination of the eigenvectors $\{u\}_r$, where $\{u\}_r$ satisfy the equations:

$$[D]\{u\}_r = \lambda_r \{u\}_r \qquad \lambda_r = \frac{1}{\omega_r^{\,2}} \qquad \text{3-15}$$

r=1,2,3,...,n

Solutions can be given an interesting interpretation in terms of linear transformations. Specifically matrix $[D]$ can be regarded as representing a linear transformation that transforms any eigenvector $\{u\}_r$ into itself, within the constant scalar multiplier λ_r . On the other hand, if an arbitrary vector $\{v\}_1$, other than eigenvector, is premultiplied by $[D]$, then the vector will be transformed into an other vector $\{v\}_2$ .Of course, the procedure can be repeated with $\{v\}_2$ as a new trialvector. A good assumption for the first trial vector end with faster and better convergence.

This iteration scheme has a major advantage in the sense that if an error is made in one of the iteration steps, this only sets back iteration process but does not affect the final result. The iteration leads to the first mode. The question remains as to how to obtain the higher modes. It shall be considered instead of another technique not suffering from this drawback , where the method is called matrix deflation.

If λ_1 and $\{u\}_1$ are the first eigenvalue and eigenvector associated with the dynamical matrix $[D]$, and $\{u\}_1$ is normalized so as to satisfy $\{u\}^T{}_1[m]\{u\}_1 = 1$ then the matrix

$$[D]_2 = [D] - \lambda_1 \{u\}_1 \{u\}_1^{T} [m] \qquad \text{3-16}$$

has the same eigenvalues as $[D]$ except that λ_1 is replaced by zero. The new iteration process using $[D]_2$ leads invariably to second mode. The process can be repeated up to enough number of modes are found.

Actually the power method using matrix deflation works also for the case of repeated eigenvalues, provided that the eigenvectors corresponding to a repeated eigenvalue are orthogonal to each other. On the other hand if eigenvectors are not computed with sufficient accuracy , deflation matrices become progressively inaccurate, thus propagating the error.

3.2.1.2 FREE VIBRATION OF THE TORSIONAL SYSTEM BY USING HOLZER METHOD

Holzer's method is essentially a trial-and-error scheme to find the natural frequencies of undamped, damped semidefinite, fixed or branched vibrating systems involving linear and angular displacements. The method can also be programmed for computer applications. A trial frequency of the system is first assumed, and a solution is found when the assumed frequency satisfies the constraints of the system. This generally requires several trials. Depending on the trial frequency used, the fundamental as well as the higher frequencies of the system can be determined. The method also gives the mode shapes.

Consider the undamped torsional semidefinite system shown in Figure 3-4. The equation of motion of the discs can be derived as follows.

$$J_1\ddot{\theta}_1 + k_{t1}(\theta_1 - \theta_2) = 0 \qquad 3\text{-}17$$

$$J_2\ddot{\theta}_2 + k_{t1}(\theta_2 - \theta_1) + k_{t2}(\theta_2 - \theta_3) = 0 \qquad 3\text{-}18$$

$$J_1\ddot{\theta}_1 + k_{t1}(\theta_3 - \theta_2) = 0 \qquad 3\text{-}19$$

Since the motion is harmonic in a natural mode of vibration, it is assumed that

$$\theta_i = \Theta_i \cos(\omega t + \phi) \qquad 3\text{-}20$$

then the following equations can be written:

$$\omega^2 J_1 \Theta_1 = k_{t1}(\Theta_1 - \Theta_2) \qquad 3\text{-}21$$

$$\omega^2 J_2 \Theta_2 = k_{t1}(\Theta_2 - \Theta_1) + k_{t2}(\Theta_2 - \Theta_3) \qquad 3\text{-}22$$

$$\omega^2 J_3 \Theta_3 = k_{t2}(\Theta_3 - \Theta_2) \qquad 3\text{-}23$$

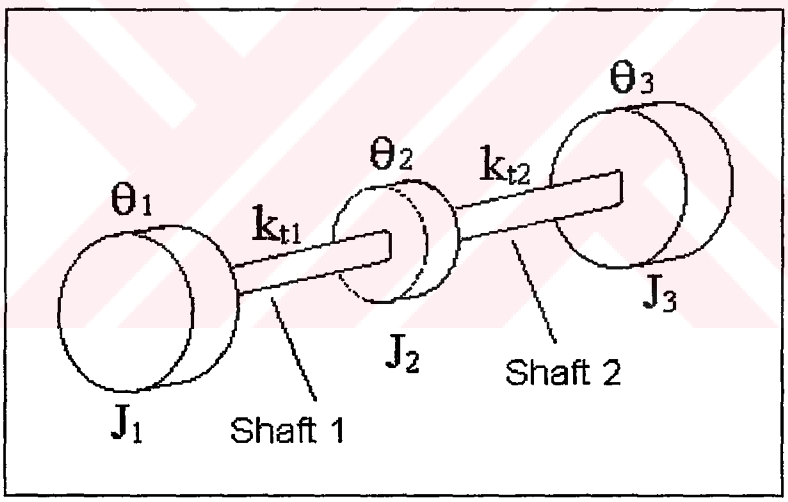


Figure 3-4 Torsional Shaft System

Summing these equations gives :

$$\sum_{i=1}^{3} \omega^2 J_i \Theta_i = 0 \qquad 3\text{-}24$$

Equation 3-24 states that the sum of inertia torques of the semidefinite system must be zero. The equation can be treated as another form of the frequency equation, and the trial frequency must satisfy this requirement.

In Holzer's method, a trial frequency ω is assumed, and Θ_1 is arbitrarily chosen as unity. Next Θ_2 is computed from Equation 3-22 and then Θ_3 is found from Equation 3-23. Thus, the following equation can be written:

$$\Theta_1 = 1 \qquad 3\text{-}25$$

$$\Theta_2 = \Theta_1 - \frac{\omega^2 - J_1\Theta_1}{k_{t1}} \qquad 3\text{-}26$$

$$\Theta_3 = \Theta_2 - \frac{\omega^2}{k_{t2}}(J_1\Theta_1 + J_2\Theta_2) \qquad 3\text{-}27$$

These values are substituted in Equation 3-24 to verify whether the constraint is satisfied, a new trial is value of ω is assumed and the process is repeated. Equations 3-24 , 3-26 and 3-27 can be generalized for an n disc system as follows :

$$\sum_{i=1}^{n} \omega^2 J_i \Theta_i = 0 \qquad 3\text{-}28$$

$$\Theta_i = \Theta_{i-1} - \frac{\omega^2}{k_{ti-1}}\left(\sum_{k=1}^{i-1} J_k \Theta_k\right), \quad \text{i=2,3,...} \qquad 3\text{-}29$$

Thus, the method uses Equation 3-29 repeatedly for different trial frequencies. If the assumed trial frequency is not a natural frequency of the system Equation 3-28 is not satisfied.

3.3 FORCED RESPONSE OF THE SYSTEM

The natural frequencies and the mode shapes of a system may give quite useful information about the response under certain excitations. The responses are the critical speeds from which a designer must certainly avoid. However, the free vibration analysis of a system is not sufficient when the magnitude of vibrations or the time history of the response are of concern. Then the forced response analysis must be performed.

In the present study, forced responses of the torsional and transverse system excitations due to unbalances and alternating forces are studied.

A rotating shaft, due to unbalanced masses and alternating forces repeating every revolution, is set to harmonic motion, i.e. vibration of the form $y_0 \cos\omega' t$, where y_0 is the vibration amplitude, variable along the shaft, and ω' is the frequency of the excitation.

If at any node there is a unbalanced mass m_i, there occurs an unbalanced forcing which is constant in magnitude but changes its direction. Since acceleration is the second derivative of displacement with respect to time, at every position along the shaft there will be an acceleration $a = -R_i\omega^2 \cos\omega t$ in magnitude where R_i is the distance between the unbalanced mass and the geometric center of the shaft and ω is the known rotational frequency. Therefore, if at any node there is a mass m, there will be an inertia force $m_i R_i \omega^2 \cos\omega t$. In fact classifying unbalanced force as an alternating force which has an amplitude of $m_i R_i \omega^2$ is not a wrong assumption for conservative design purpose. On the other hand, alternating forces may act with a frequency other than the operational speed of shaft.

Finding the maximum values of the displacements due to various alternating forces is the main problem of this study. For example, if there are alternating forces acting with different frequencies ($\omega, \omega_1, \omega_2, \ldots \omega_n$), maximum values of the displacement at a node will be the sum of their absolute values ($|y| + |y_1| + |y_2| + \ldots |y_n|$).

3.3.1 HARMONIC EXCITATION

To obtain the solution of forced response of the system, the eigenvalue problem should be solved first. After finding the eigenvalues and eigenvectors, it is very straight forward to find the excitations due to alternating forces which was assumed to be harmonic.

After finding all of the eigenvalues and eigenvectors, the following definition can be made:

$$\{q\}=[\Phi]\{\eta\} \tag{3-30}$$

where $\{\eta\}$ is the response according to principal coordinates and $[\Phi]$ is the normalized eigenvectors so as to satisfy :

$$[\Phi]^T[M][\Phi]=[I] \tag{3-31}$$

Introducing Equation 3-30 into equation of motion, premultiplying the result by $[\Phi]^T$ the following equation can be obtained.

$$\{\eta''\}+\omega^2\{\eta\}=\{P\} \tag{3-32}$$

where

$$\{P\}=[\Phi]^T\{F\} \tag{3-33}$$

If this differential equation is solved for the r-th mode, harmonic excitations can be found by the following equation.

$$\eta_r = \frac{P_r}{\omega_r^2 - \omega^2} e^{i\omega t} \qquad \text{for } r=1,2,3...n \qquad 3\text{-}34$$

Then, the responses that are found according to the principal coordinate system should be converted to our coordinate system that have been chosen by using the Equation 3-30.

In section (3.2.1.1) it is mentioned that finding all of the eigenvalues of the system is very difficult to converge and takes much processing time. However, Equation 3-34 shows that it is possible to truncate the higher modes of the system as for higher modes, $1/(\omega_r^2 - \omega^2)$ is getting smaller and is negligible.

CHAPTER 4

OPTIMIZATION

In the pervious chapters, criteria to design power transmission shaft is selected and the constraints are determined so that it will perform its duty satisfactorily without failure. The disadvantages of the foregoing procedure is obvious. The design can be highly uneconomical even when an intuitively selected solution satisfies the behavioral constraints. So that is not enough for the goal associated with the model. A goal of every designer is to design the best systems.

It is important to contrast the optimum design process with conventional design process. Figure 4-1 summarizes basic steps that are common to both processes, the optimum design process differs from the conventional design process in the following ways.

The conventional design process for the power transmission shaft design problem is to select some initial values for the design variables. Initial guess of variables are done by design procedure where variable are assumed to be dependent on each other with fillet radii and the minimum wall thickness according to the sign convention shown in the Figure 4-2:

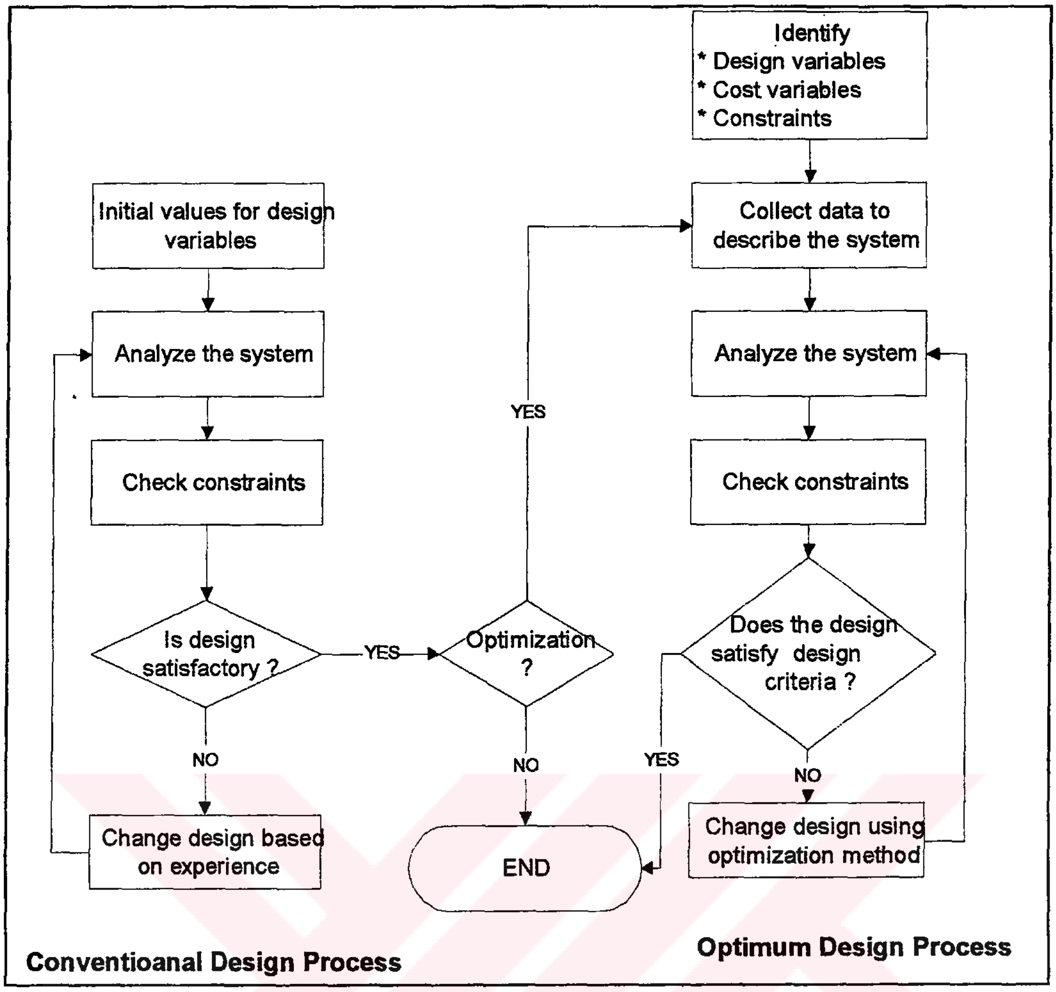


Figure 4-1 Conventional Design Versus Optimum Design Process

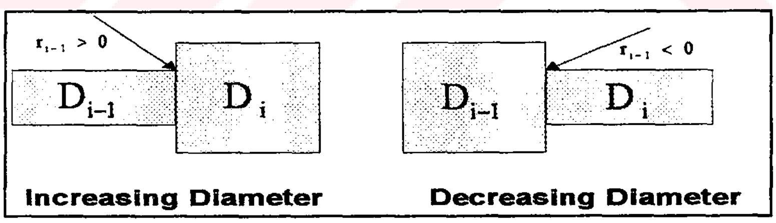


Figure 4-2 Sign Convention of Fillet Radii

$$D_i = D_{i-1} + 2r_{i-1} \tag{4-1}$$

$$D_0 = D_1 - 2w_{tick}\ , \quad \text{if } D_1 \le D_n \tag{4-2}$$

$$D_0 = D_n - 2w_{tick}\ , \quad \text{if } D_1 > D_n \tag{4-3}$$

If one of the constraints is exceeded, then design variables are adjusted and process is repeated until the variables are in the allowable limit. Next, the other constraints are checked until an acceptable design is obtained. In contrast, the optimum design process takes into account all the constraints simultaneously, and iteratively improves the design, while minimizing the cost function that can result in better designs. New solutions can be obtained very rapidly with optimum design process once the problem has been formulated and coupled to an optimization program.

It can be seen that optimum design process is more formal than the conventional design process. The designer must clearly and precisely identify design variables, a design performance measure (the cost function) by which the design satisfies its optimum, and all the performance constraints that the final design must satisfy.

4.1 FORMULATION OF THE OPTIMIZATION PROBLEM

It is desirable and more economical to have a minimum weight which would satisfy the constraint condition requirements in most design problems. For shaft design, it is required to minimize the weight of shafts which would satisfy the strength, fundamental critical speed and deflection allowed.

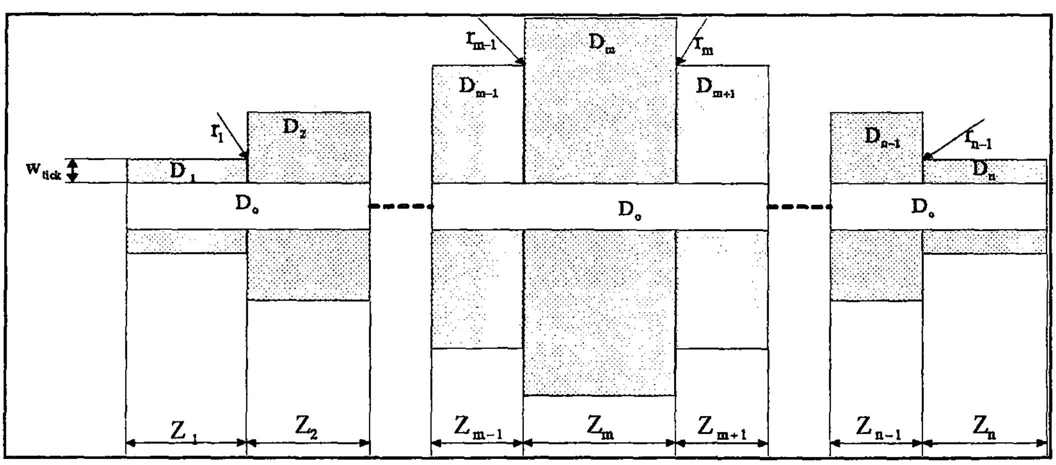


Figure 4-3 Sample Shaft to be Optimized

In Figure 4-3 D_i , i=0,1,2,3,... ,n are the independent variables. Then the objective of shaft optimization problem can be formulated as :

$$F(D) = \frac{\pi \rho_s}{4}\left[(Z_1 D_1^3 + Z_2 D_2^3 + Z_3 D_3^3 + ...Z_n D_n^3) - (Z_1 + Z_2 + ...Z_n) D_0^3\right] \quad 4\text{-}4$$

D_i : Diameter at the segments

(D_0 corresponds to inside diameter)

Z_i : Shaft length starting from one segment to the other

r_i : Fillet radii

ρ_s : Density of steel , $76 \times 10^3 \ N/m^3$

If this objective function is only subjected to the design constraints which are implicit variables, determined in the previous chapter, meaningless solutions may be obtained. So some explicit constraints that can be called physical constraints should be added. Although design variables D_i are seemed to be independendent of each other in the objective function, they are dependent on some physical inequalities, such as:

$$D_{i-1} + 2r_{i-1} \leq D_i \leq D_{i+1} - 2r_i \quad ; \text{ for } i=2,3,....m-1 \quad 4\text{-}5$$

$$D_{i+1} - 2r_i \leq D_i \leq D_{i-1} + 2r_{i-1} \quad ; \text{ for } i=m+1,m+2,....n-1 \quad 4\text{-}6$$

$$D_{m-1} + 2r_{m-1} \leq D_m \leq D_{\max} \quad ; \text{ if } D_{m-1} \geq D_{m+1} \quad 4\text{-}7$$

$$D_{m+1} - 2r_m \leq D_m \leq D_{\max} \quad ; \text{ if } D_{m-1} < D_{m+1} \quad 4\text{-}8$$

for the case of hollow shafts ;

$$0 \le D_0 \le D_1 - 2w_{tick} \quad ; \text{ if } D_1 \le D_n \tag{4-9}$$

$$0 \le D_0 \le D_n - 2w_{tick} \quad ; \text{ if } D_1 > D_n \tag{4-10}$$

$$D_0 + 2w_{tick} \le D_1 \le D_2 - 2r_i \tag{4-11}$$

$$D_0 + 2w_{tick} \le D_n \le D_{n-1} + 2r_{n-1} \tag{4-12}$$

for the case of solid shafts ;

$$D_0 = 0 \tag{4-13}$$

$$D_{\min} \le D_1 \le D_2 - 2r_i \tag{4-14}$$

$$D_{\min} \le D_n \le D_{n-1} + 2r_{n-1} \tag{4-15}$$

It is seen that the diameters at the sections are constrained with their neighborhood geometries. However, D_m and the end conditions are constrained with $D_{\min}, w_{tick}$ and $D_{\max}$ which depends on the users decisions. The definition of D_m is obvious. It is the maximum diameter of the sections at the index m where user defined while entering the notchcodes and the fillet radii. Optimization procedure changes its value, but never changes the index.

In contrast design constraints are dependent functions of the explicit independent variables.

$$N \le N_c(D_0, D_i) \tag{4-16}$$

$$\omega \le \omega_c(D_0, D_i) \tag{4-17}$$

$$S_{l\max} \ge S_{lc}(D_0, D_i) \tag{4-18}$$

$$Y_{l\max} \geq Y_{lc}(D_0, D_i) \quad 4\text{-}19$$

$$Y_{a\max} \geq Y_{ac}(D_0, D_i) \quad 4\text{-}20$$

where

N	: Minimum allowable factor of safety
N_c	: Minimum factor of safety calculated at the critical section
ω	: Operational speed of shaft
ω_c	: Fundamental critical speed
S_{lc} , Y_{lc}	: Maximum lateral slope and deflection calculated at the critical section
$S_{l\max}$, $Y_{l\max}$	: Maximum allowable lateral slope and deflection
Y_{ac}	: Maximum angular deflection calculated at the critical section
$Y_{a\max}$	: Maximum allowable angular deflection

4.2 CONSTRAINED ROSENBROCK ALGORITHM

This method is a sequential search technique which has proven effective in solving some problems where the variables are constrained

Optimize $F(D_0,D_1,D_2,...,D_n)$

Subject to $G_k \le D_k \le H_k$, k=1,2,......NOC

where

G_k : the lower constraints

H_k : the upper constraints

NOC: number of constraints

The procedure requires a starting point that satisfies the constraints and does not lie in the boundary zone. In order to guarantee this , instead of guessing initial values , program collect them from the conventional design procedure as shown in the Figure 4-1. Then the following steps are carried out.

- Define by F^0 the current best objective function value for a point where the constraints are satisfied, and F^* the current best objective function value for a point where the constraints are satisfied and in addition the boundary zones are not violated. They are initially set equal to the objective function value at the starting point.

- If the current point objective function evaluation, F, is worse than F^0 or if the constraints are violated, the trial is a failure and the unconstrained procedure is continued.

- If the current point lies within a boundary zone, the objective function is modified as follows :

$$F_{new} = F_{old} - (F_{old} - F^*)(3\lambda - 4\lambda^2 + 2\lambda^3) \qquad 4\text{-}21$$

where

$$\lambda = \frac{[\,G_k + (H_k - G_k)10^{-4}\,] - D_k}{(H_k - G_k)10^{-4}} \quad \text{(lower zone)} \qquad 4\text{-}22$$

$$\lambda = \frac{D_k - [H_k - (H_k - G_k)10^{-4}\,]}{(H_k - G_k)10^{-4}} \quad \text{(upper zone)} \qquad 4\text{-}23$$

  Thus, the function value is replaced by the best current function value in the feasible region and not in a boundary zone.

- If an improvement in the objective function has been obtained without violating the boundary zones or constraints, F^* is set equal to F^0 and procedure continued.

- The search procedure is terminated when the convergence criteria is satisfied.

A flowchart showing the Rosenbrock procedure is given in Figure 4-4.

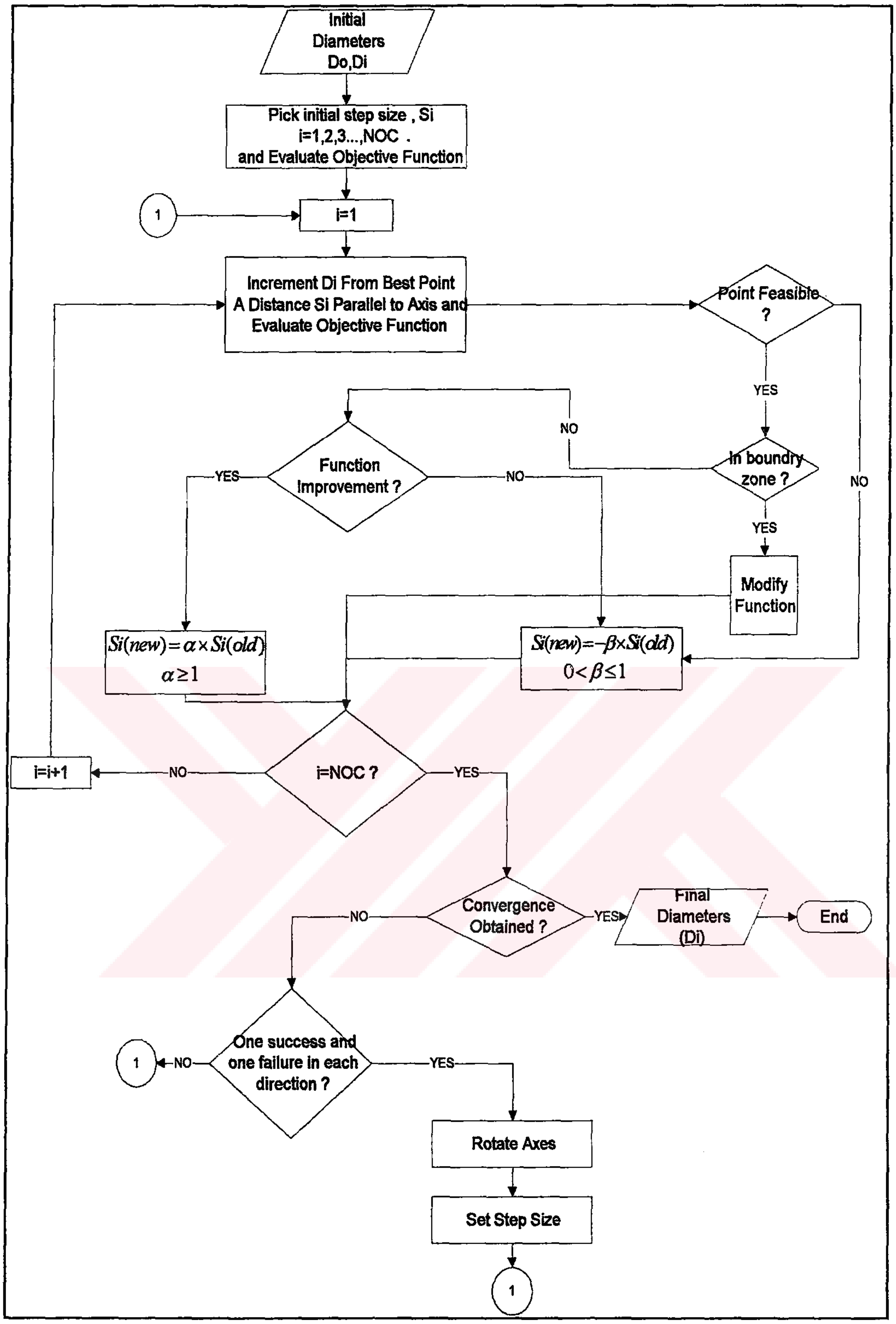


Figure 4-4 Flowchart of the Rosenbrock Algorithm

CHAPTER 5

COMPUTER PROGRAM

In this chapter, the structure and the description of the implementation features of the developed program will be discussed in detail.

The program is composed of some subsystems which we call functional modules. The modules have different priority levels within the system. Since the other machine elements are connected to the shaft , the shaft module is the backbone of the system. Each functional module has a number of design factors. These factors are classified as design variables and performance characteristics . Some of the design factors are shared by more then one functional module. They are mainly performace characteristics or general input specifications to the system.

The developed program is a part of a whole Design of Machine Elements Project. In Figure 5-1, general approach to the problem is presented. The development of shaft module is the main scope of this study. The interaction between the main modules of the project is obtained using the same input- output files.

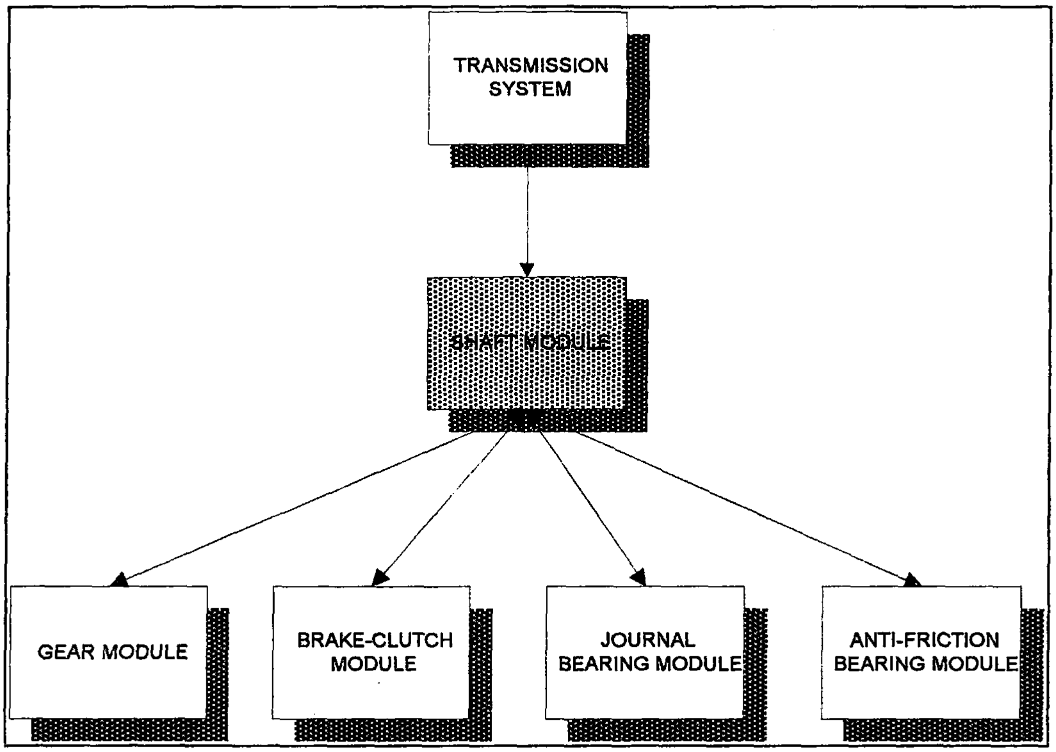


Figure 5-1 Interactions Between the Main Parts of the Project

The program mainly consists of two modules , Visual-Basic and C modules namely. Visual Basic is one of the most commonly used windows based programming language that supplies powerful user interface. On the other hand C is a general programming language, however , it is not restricted to any operating system. Although C matches the capability of many computers, it is independent of any particular machine architecture. With a little care it is easy to write portable programs, that is , programs that can be run without change on a variety of hardware. This means that , using ANSI C standards the same code can be run under UNIX , MACOS, DOS , WINDOWS 3.1 , WINDOWS 95 , ... etc. environments. This property is unique to C language and with the help of this feature the main goals of the project is satisfied which is modularity ,flexibility and multienvironment support.

5.1 THE STRUCTURE OF THE COMPUTER AIDED DESIGN OF POWER TRANSMISSION PROJECT

Computer Aided Design of Power Transmission Project consists of several design modules . As shown in Figure 5-2 each of these modules are the subject of several master thesis those are being done in Mechanical Engineering Department of METU.

Each module develops a design according to the input data that are requested from the other project . For example, in this thesis a shaft is designed according to the transmission system constraints and the specifications of the gears , bearings,... etc. on the shaft.

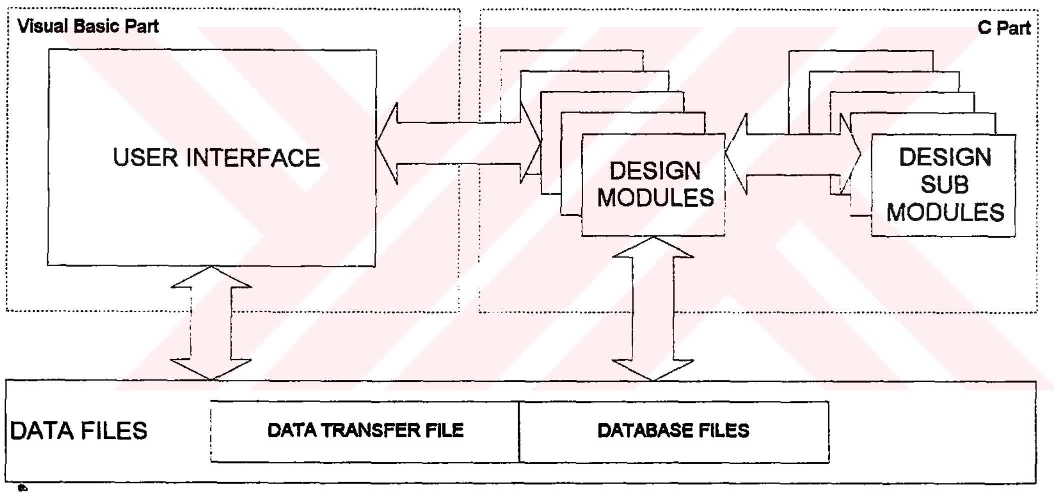


Figure 5-2 The Complete Design Project

User interface part takes necessary data from user and generates a data file that is the input file for the design calculation module. Design module calculates the required results by the help of design sub modules and database files if necessary. The obtained results are stored in an output file. This file is read again by the user interface part and shown as the result in appropriate way .

5.2 THE STRUCTURE OF THE SHAFT DESIGN PROGRAM

The structure of the shaft design program is developed so that it will be proper to the system design structure mentioned in the previous section.

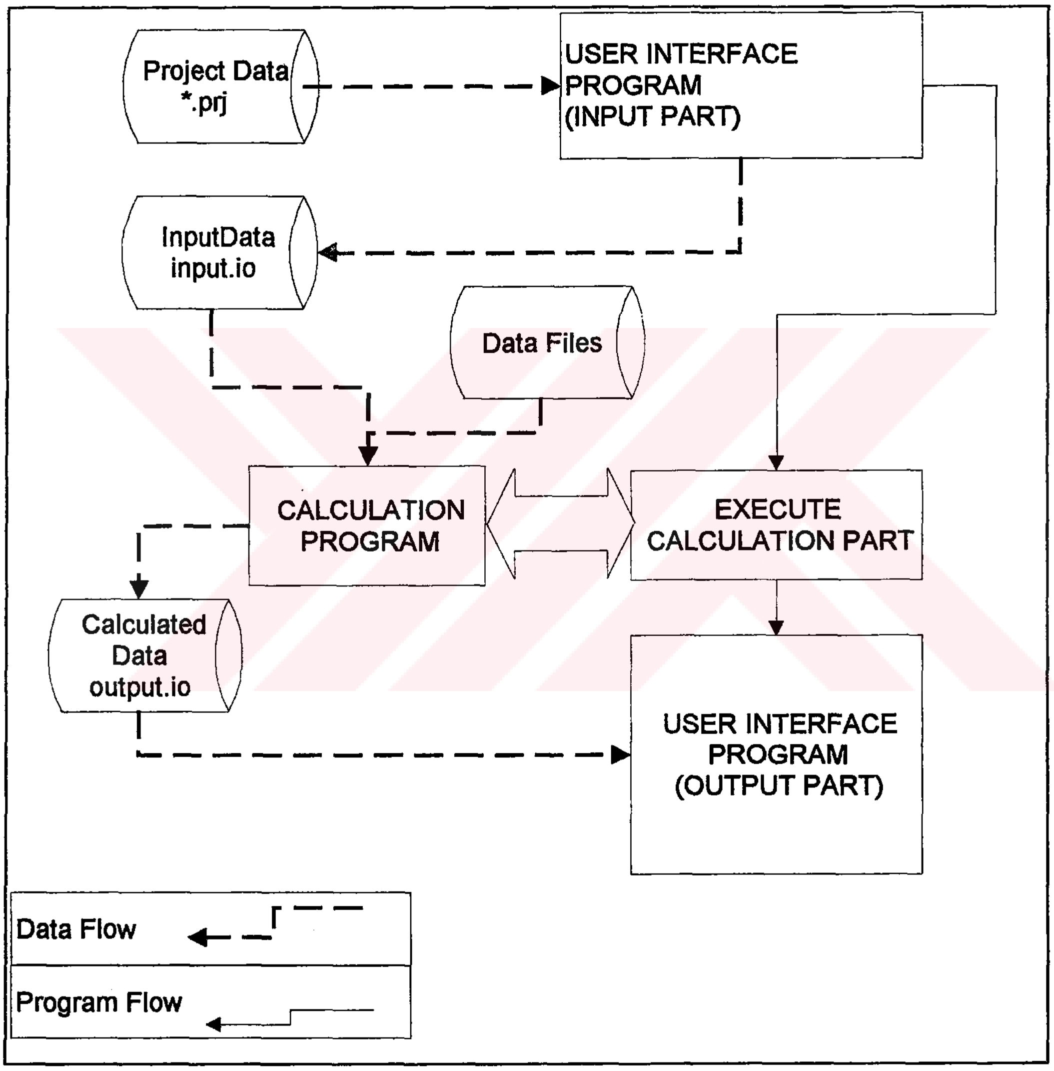


Figure 5-3 The Structure of the Shaft Design Program

In the first step, user can start a previous project or create a new project. If user starts a previous project, the stored design conditions are placed in their fields. Then program is ready to accept design conditions.

After the reception of the inputs, program stores the input data in a file "input.io", then the execution of the calculation part starts as described in Figure 5-3. Calculation program reads the input information from the "input.io". If the calculation needs some information other than inputs "Data Files" takes its place. The data is extrapolated or interpolated by the program if it is needed. Before the calculation part comes to an end, it outputs the results to the file "output.io" and program returns back to User Interface for representation of output including comparison of different design features (Table and graphical).

5.2.1 USER INTERFACE MODULE

User interface part of the program is developed by using Visual Basic 4.0 in MS-Windows environment and consist of 15 forms and a help file. The program is driven by pull down menu system which makes interface user-friendly. The flowchart of the User Interface of the program is given in APPENDIX A. Input Menu Structure of the Program is given in Figure 5-4.

User interface part aims three important target

- Graphical input and graphical visualization of input data
- Representation of output including comparison of different design features
- Reporting

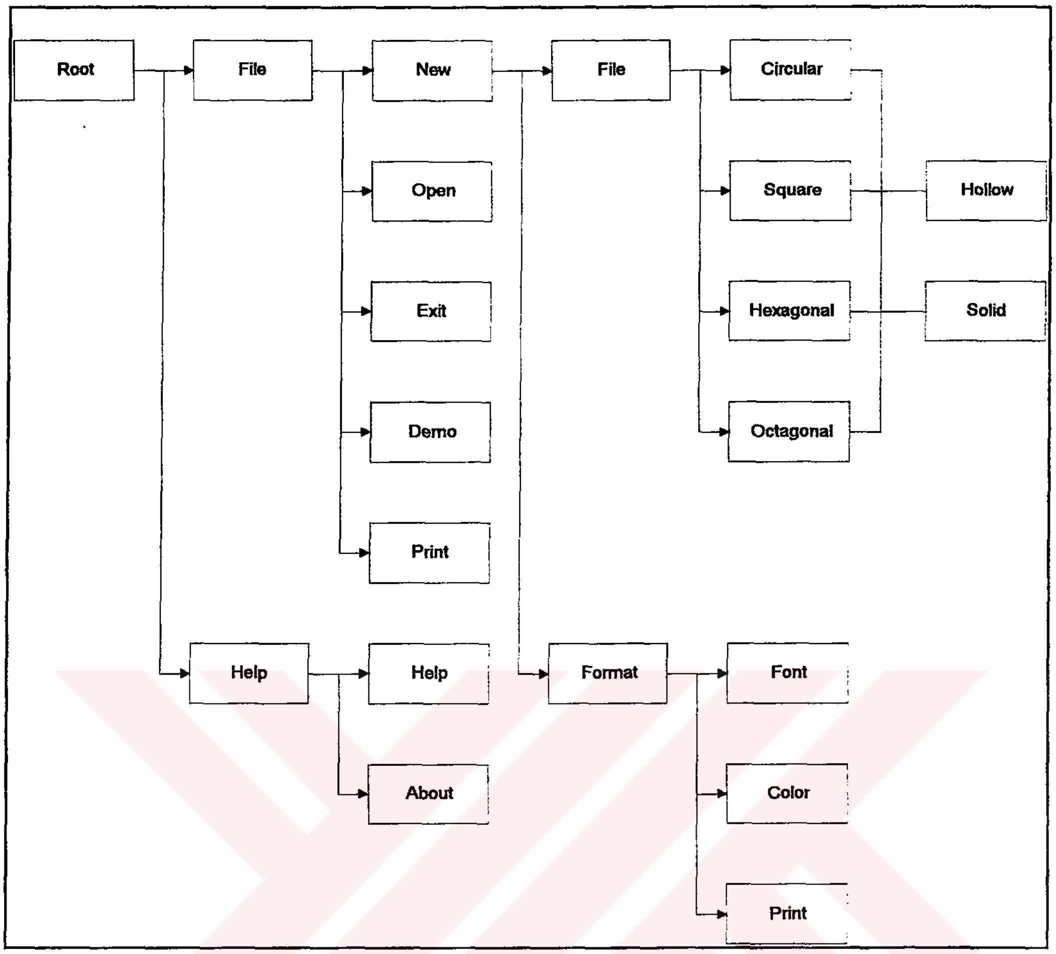


Figure 5-4 Input Menu Structure of the Program

5.2.1.1 THE STRUCTURE AND USE OF DATA FILES

The following data files are used for different purposes at different stages of the software.

- record data files
- data transfer files
- database files

Since the program is integrated between User Interface and Calculation Parts, data transfer between these parts are made with two files which are "input.io" and "output.io". Information about the input that the user enters are kept in the "input.io" and the results of the calculation is stored "output.io". Record data files are the copy of "input.io" which gives the designer the capability of opening and saving input information. These files are named "*.prj". Other data information that is needed in calculating the results are kept in database files which are the submodules of C compiler.

5.2.2 MAIN CALCULATION PROGRAM

Main calculation part of the program is developed by using TC 3.0 in MS-DOS environment consists of Design Sub-Programs (*.h) and a main program (*.cpp) file. Calculation program reads "input.io" and table files "Table.H" for inputs and produces "output.io" as output. The files that are used in the calculation program are:

Shaft.cpp : Determines the order in design submodules.

Input.h : Reads the input information from the data transfer file "input.io".

Output.h : Writes the results to the data transfer file "output.io"

Eigen.h : Calculates the eigenvalues and eigenvectors of the matrices.

Force.h : Calculates reaction forces at the bearings and shearing forces , bending moments , torque and axial forces at the stations.

Def.h : Calculates static deflections at the stations.

Endurenc.h : Calculates endurance limit of the material.

Lump.h : Divide the shaft in to lump masses and calculates critical speeds.

Strength.h : Calculates stresses and factor of safety at the stations.

Matrix.h : Makes some matrix manipulations.

Mikro.h : Sub-module in which some of the simple functions or the constants are defined.

Holz .h : Finds eigenvalues and eigenvectors for torsional system.

Response.h : Finds the dynamic deflections and the slopes at the stations.

Opt.h : Iterates for optimization.

Table.h : Extrapolates or interpolates some of the stored data like stress concentration factors that are needed for calculations.

The Design Sub-programs such as Matrix.h can be used by the other design programs whenever needed. This approach make the project more flexible and modular which is one of the objects of the present study.

5.3 DISCUSSION OF COMPUTER PROGRAMMING LANGUAGES

Although using two compilers like Visual Basic and C++ which are really different programming languages seems a little bit odd , this kind of integration has some benefits. These two compilers are not only different in programming language but also authoritative in different platforms. While programming the application, it is very convenient to use the advantages of these two compilers where they are powerful in. For example, you can use modularity , flexibility of C++ and simplicity in creating graphics of Visual Basic in one aplication.

CHAPTER 6

NUMERICAL RESULTS AND RELIABILITY OF THE COMPUTER PROGRAM

In this chapter some of the results obtained from the developed program will be checked and the reliability of the developed program will be investigated.

In the first part static lateral deflection calculations will be analyzed. Comparing the results with the exact values that are found by hand calculation will also give a brief idea about the accuracy in computation of reaction forces, shearing forces and bending moments. If the method is decided to be accurate enough, the static deflections will be compared with dynamic responses at a very small shaft speed where these two results must satisfy each other. Later some of the parameters such as shaft speed and bearing stiffness will be changed to examine free and forced responses. These results are discussed as if they are expected .

For testing system namely System A is examined (Figure 6-1). The parameters of the system and the applied load are listed in Table 6-1 and in Table 6-2.

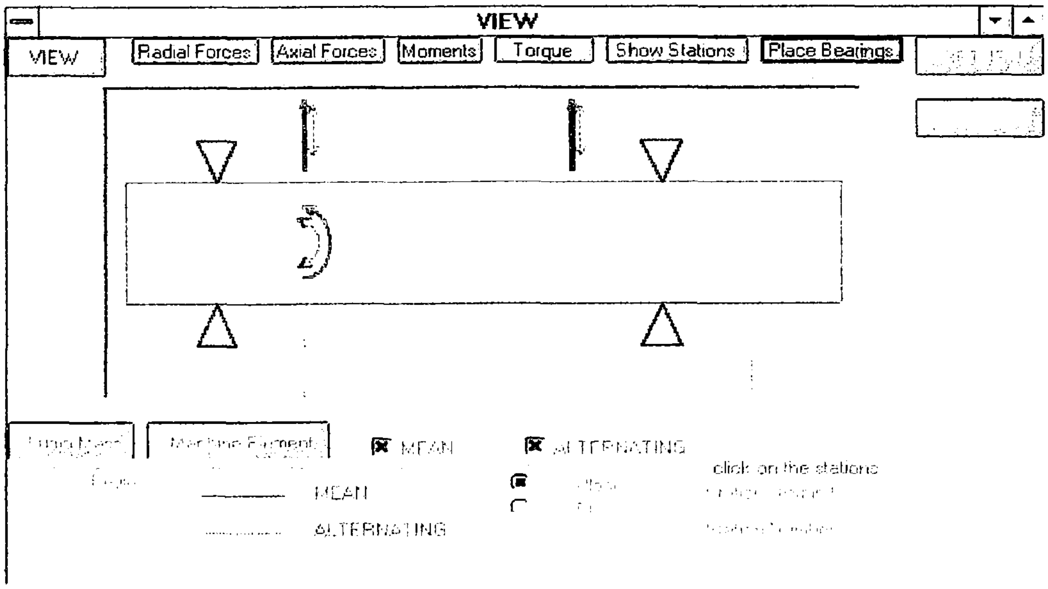


Figure 6-1 System A

Table 6-1 Parameters of System A

PARAMETES	NUMERICAL VALUES	
d shaft (set constant)=	30	mm
Number of station =	9	
Station number at first support=	2	
Station number at second support=	7	
Dist DX =	50	mm
Force at station 3 =	900	N
Force at station 6 =	900	N
Moment at station 3 =	400	Nm
Bearing type	Rigid	
Shaft type	Circular Solid	

Table 6-2 Applied Load with its Station Numbers and Distances

ST #	DIST DX (mm)	X-Y plane FORCE (mean) (N)	X-Y plane MOMENT (mean) (Nm)	X-Y plane FORCE (alternating) (N)	X-Y plane MOMENT (alternating) (Nm)	Alternating Load Type
1	50	0	0	0	0	
2	50	0	0	0	0	
3	50	900	400	900	400	$\cos(\omega_1 t)$
4	50	0	0	0	0	
5	50	0	0	0	0	
6	50	900	0	900	0	$\cos(\omega_1 t)$
7	50	0	0	0	0	
8	50	0	0	0	0	
9	50	0	0	0	0	

6.1.1 ERROR ACCUMULATION IN STATIC AND DYNAMIC DEFLECTION

System A is used in examining the accumulation of error in computing the deflection in the developed program. Computed values are compared with the results calculated using the beam formulas for shaft deformations which are given in Shigley [3]. These formulations adapted to Excel Worksheet where they can be accepted as real values. Numerical results are presented in Table 6-3.

Table 6-3 Error Accumulation For Static Deflection

STATION #	STATIC DEFLECTION(mm E3)		% ERROR
	Excel	Dev. Prog.	
1	-120.5481853	-120.5482	-1.22151E-05
2	0	0	0
3	122.3209527	122.321	-3.86643E-05
4	196.7771848	196.7772	-7.73104E-06
5	184.6210653	184.6211	-1.8815E-05
6	110.1648332	110.1649	-6.06528E-05
7	0	0	0
8	-116.4961454	-116.4961	3.90002E-05
9	-232.9922909	-232.9922	3.90002E-05

The same procedure can be used to understand the reliability in finding the dynamic deflection calculation. In order to do so, operating speed of shaft can be taken low compared to critical speed of shaft.

Table 6-4 Error Accumulation For Dynamic Deflection

STATION #	DYNAMIC DEFLECTION(mm E3)		% ERROR
	Excel	Dev. Prog.	
1	-120.5481853	-119.8418	0.585977527
2	0	0	0
3	122.3209527	123.2028	-0.72092906
4	196.7771848	195.9797	0.405272993
5	184.6210653	185.5555	-0.506136575
6	110.1648332	109.3728	0.718952826
7	0	0	0
8	-116.4961454	-116.7992	-0.260141282
9	-232.9922909	-225.402	3.257743352

Examining the above tables, it is decided that dynamic deflections are computed with less accuracy. This is expected since the static deflection algorithm is operated in finding the flexibility matrix. Other errors are collected in converging and truncating the eigenvalues and eigenvectors. On the other hand the error accumulated is reasonable and reliable.

6.1.2 BEARING STIFFNESS VERSUS COMPUTED CRITICAL SPEED

System A is used in computing the fundamental critical speed. In the system analyzed, stiffness coefficients in both of the bearings (k_1,k_2) are the same. The change in fundamental critical speeds due to various stiffness coefficients is plotted. This research is repeated by increasing the station distances to observe the effects on various shaft lengths. The graphical results calculated by the developed program is presented in Figure 6-2.

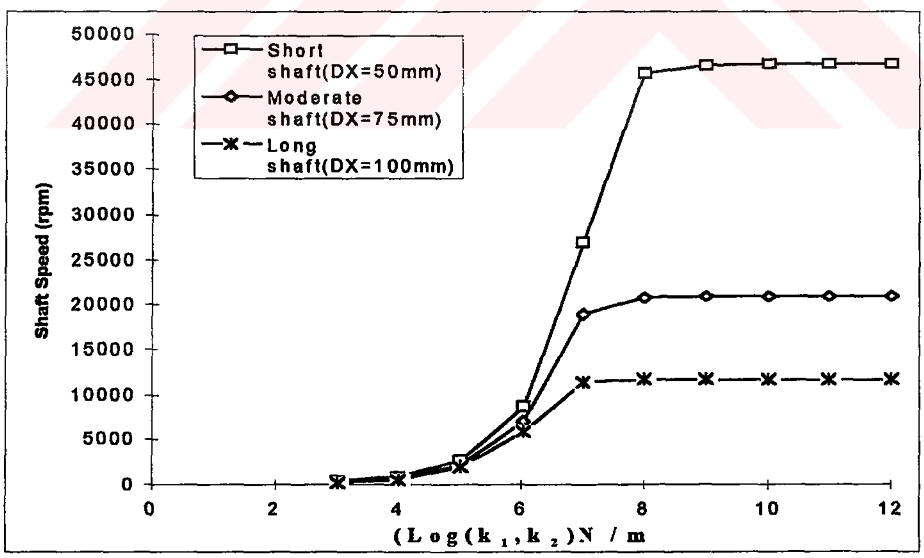


Figure 6-2 Computed Graphical Results of Bearing Stiffness on Free Response

From the above figure, it is natural to conclude that increasing bearing stiffness will increase the critical speed of the system. As seen in the Figure 6-2, computed effects of bearing stiffness on free response for three different shaft lengths are not the same. For certain range of bearing stiffness, the rate of increase in fundamental critical speed is high. Also the graph shows that , flexible bearings have more effects on free response of the system with short shaft than the system with long shaft.

Although the model that is used in the developed program is not exact enough to examine the effects of stiffness coefficients on free response, comparing the concluded result with the study of Okan [19], it shows that the model is effective.

6.1.3 OPERATING SPEED OF SHAFT VERSUS COMPUTED FORCED RESPONSE

In section (6.1.1) it is decided that program works reasonable if the operating shaft speed is very low compared with critical speed. Hence, developed program must find dynamic responses at higher shaft speeds in order to constraint the design for deflection.

The efficiency of the developed program in finding forced responses is tested under number of operating speeds by using System A. The results are plotted in Figure 6-3.

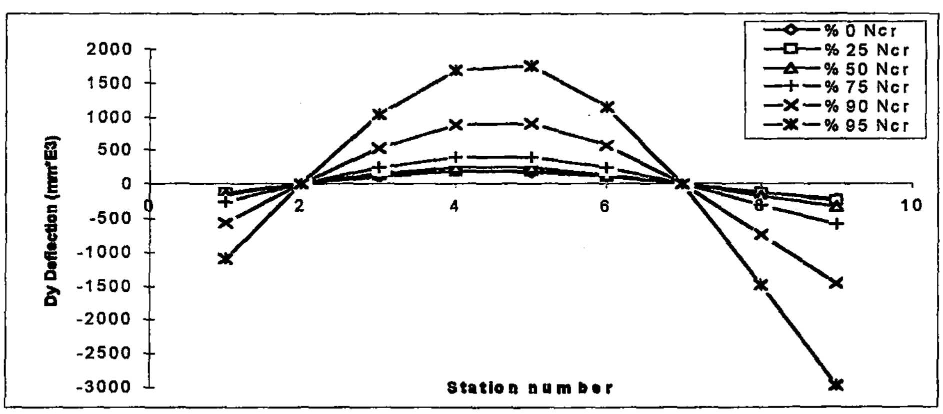


Figure 6-3 Graphical Results of Forced Response Under Different Shaft Speeds

Although it is not very accurate to examine the effect of operating speed of shafts on forced response due to the vibration model that is implemented to the developed program, above diagram shows that operating speed of shaft will increase the dynamic deflection of the system. It is an approval of the known facts that are given in literature and also verifies the program does not fail while analyzing these effects.

CHAPTER 7

CASE STUDIES AND DISCUSSION OF RESULTS

To show the capabilities of the developed software some case studies will be considered.

As an example a sample shaft shown in Figure 7-1 is used for the parameters given in Table 7-1. Table 7-2 shows the number of stations and loads acting on X-Y plane and X-Z plane. Numerical values of these loads are given in Table 7-2.

As a first case, the sample will be examined under "Analyze", "Conventional Design" and "Optimization" process of the developed program. Advantages and disadvantages between "Conventional Design" and "Optimization" process will be discussed.

Later new parameters like alternating loads will be added to the problem. This case will be tried under the "analyze" process. Differences between the static and dynamic responses will be compared and changes in constraints will be verified with the results of the first case.

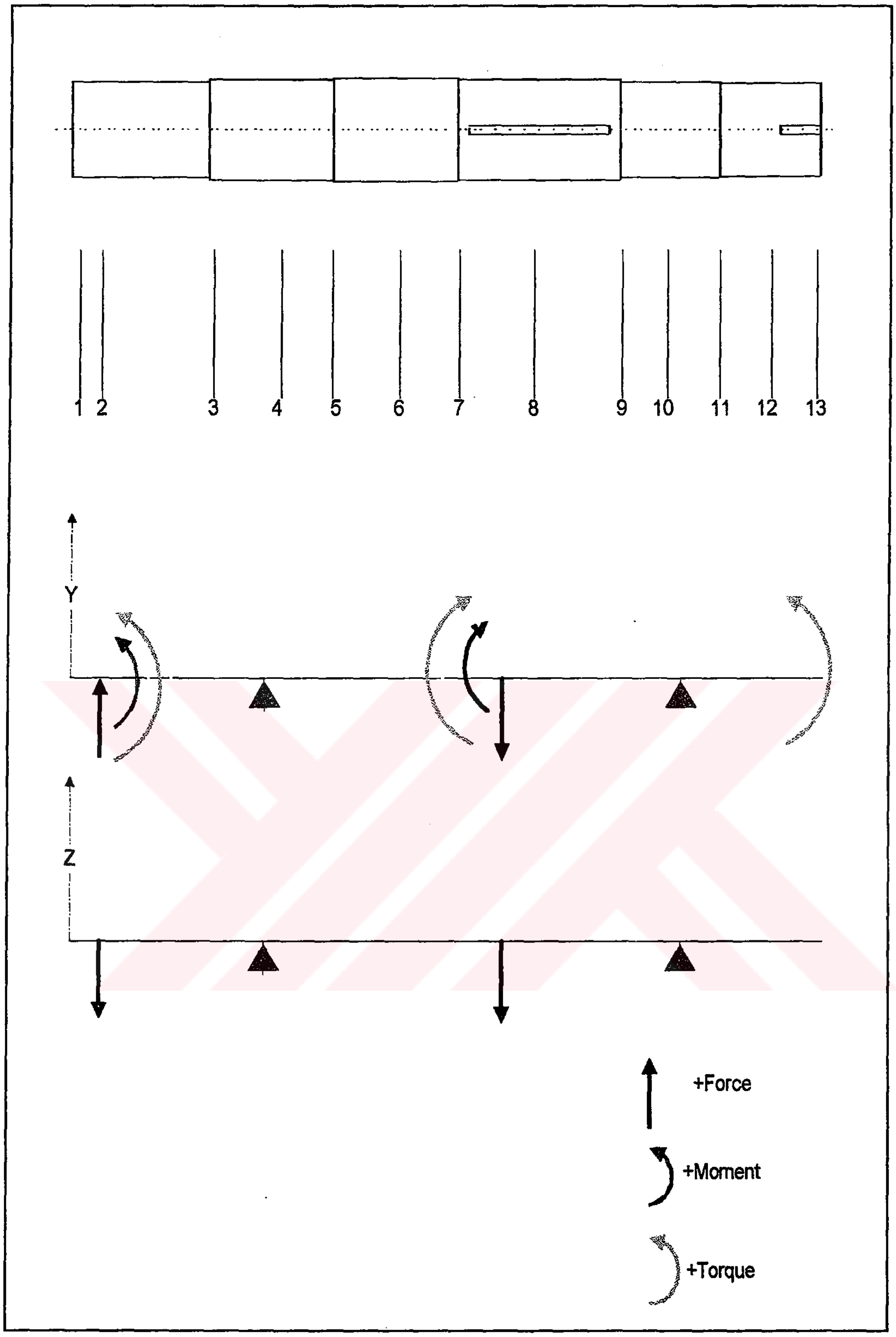


Figure 7-1 Schematic Representation of the Sample Shaft with its Station Numbers and Applied Loads

Table 7-1 Input Information of the Sample Shaft

Type of the Shaft	:Solid Circular Stepped Shaft
Surface Condition	: Ground
Maximum Temperature	: 100 C
Reliability	: 99 %
Factor of Safety	: 3.0
Material	: AISI 1040 CD
Tensile Strength	: 689 MPa
Yield Strength	: 606 MPa
BHN	:207
Shaft Speed	:1500 rpm
Design Approach	:Soderberg
Number of Station	:13
Station Number of First Support	: 4
Station Number of Second Support	:10
Bearing Stiffness of First Support	: Rigid
Bearing Stiffness of Second Support	: Rigid
Keyway of Profile Type at Station 8	: Sled-runner
Keyway of Profile Type at Station 13	: Sled-runner

Table 7-2 Applied Loads with its Station Numbers and Distances

ST #	DIST DX (mm)	NOTCH	NOTCH RAD (mm)	X-Y plane FORCE (mean) (N)	X-Z plane FORCE (mean) (N)	X-Y plane MOMENT (mean) (Nm)	TORQUE (mean) (Nm)
1	0	Smooth	0	0	0	0	0
2	30	Smooth	0	3000	-2500	500	300
3	80	Incr. Dia.	0.2	0	0	0	0
4	50	Smooth	0	0	0	0	0
5	50	Incr. Dia.	0.2	0	0	0	0
6	50	Smooth	0	0	0	0	0
7	50	Decr. Dia.	-0.2	0	0	0	0
8	50	Keyway	0	-2000	-3500	-300	-250
9	80	Decr. Dia.	-0.2	0	0	0	0
10	40	Smooth	0	0	0	0	0
11	40	Decr. Dia.	-0.2	0	0	0	0
12	40	Smooth	0	0	0	0	0
13	40	Keyway	0	0	0	0	-50

CASE 1-A : In the first run of the program , shaft is analyzed for the design criteria which are strength, rigidity and stability in favor of the given parameters by entering a constant diameter of 50 mm for the first section of the shaft. Diameters for the other sections are found by the program dependently and automatically by adding fillet radii as it is mentioned in Chapter(2).

Numerical results calculated by the developed program are given in Table 7-3 and Table 7-4. Graphical outputs of these results are presented in Figure 7-2 -Figure 7-7.

Table 7-3 Numerical Results of the Loading for X-Y Plane

ST #	XY-Plane SHEAR Nm	X-Y Plane BENDING Nm	X-Y Plane SLOPE deg*10E-3	X-Y Plane DEF mm*(10*E-3)
1	0	0	44.1	-73.9574
2	0	0	44.1	-50.8792
3	3000	-260	16.6414	-10.5113
4	3000	-110	8.5529	0
5	156.25	-102.1875	3.9096	5.4107
6	156.25	-94.375	-0.2522	6.9819
7	156.25	-86.5625	-4.0873	5.0649
8	156.25	-78.75	-7.7045	-0.1016
9	-1843.75	73.75	2.6197	-2.4478
10	-1843.75	0	3.9497	0
11	0	0	3.9497	2.7576
12	0	0	3.9497	5.5152
13	0	0	3.9497	8.2727

Table 7-4 Numerical Results of the Loading for X-Z Plane

ST #	X-Z Plane SHEAR Nm	X-Z Plane BENDING Nm	X-Z Plane SLOPE deg10E-3	X-Z Plane DEF mm*(10*E-3)
1	0	0	31.9987	-75.047
2	0	0	31.9987	-58.2996
3	-2500	-200	24.7758	-17.0008
4	-2500	-325	13.2936	0
5	2328.125	-208.5938	1.628	6.1369
6	2328.125	-92.1875	-4.7465	4.4167
7	2328.125	24.2188	-6.1854	-0.711
8	2328.125	140.625	-2.5854	-4.9061
9	-1171.875	46.875	3.9783	-3.1696
10	-1171.875	0	4.8268	0
11	0	0	4.8268	3.3666
12	0	0	4.8268	6.7331
13	0	0	4.8268	10.0997

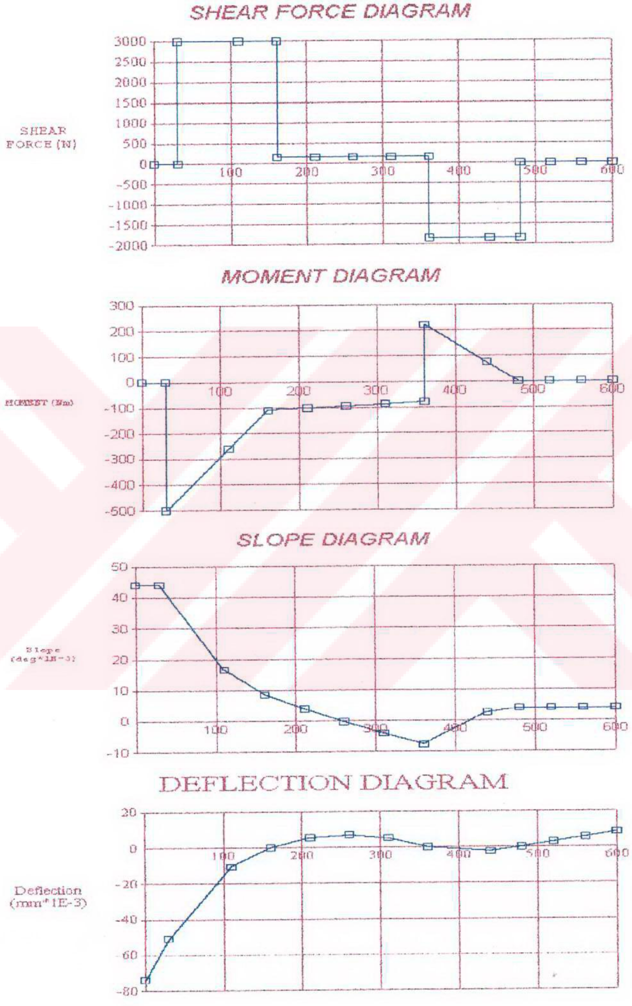


Figure 7-2 X-Y Plane Loading Diagrams

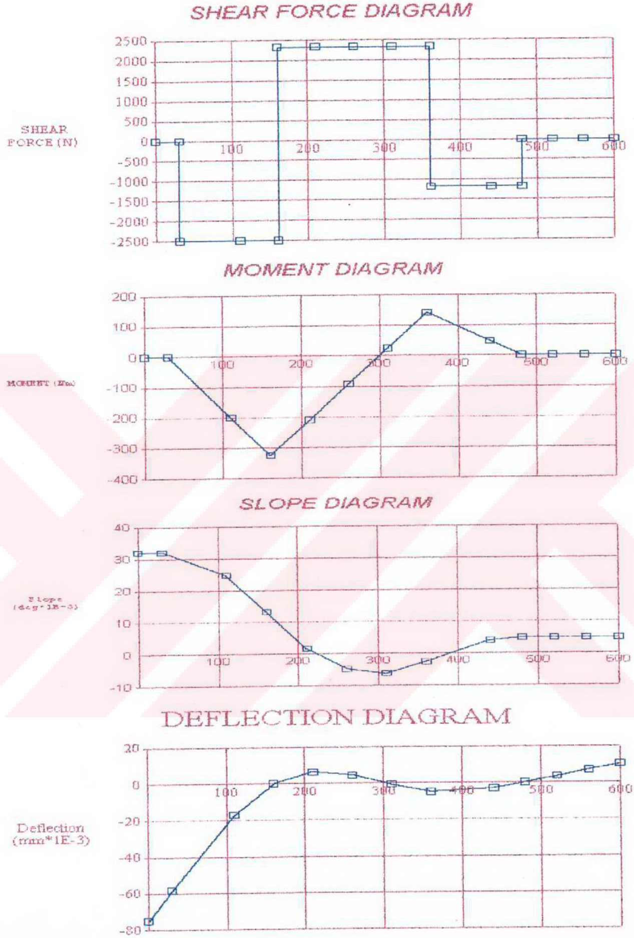
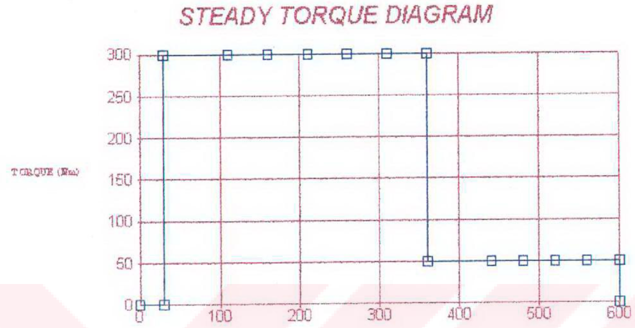


Figure 7-3 X-Z Plane Loading Diagrams



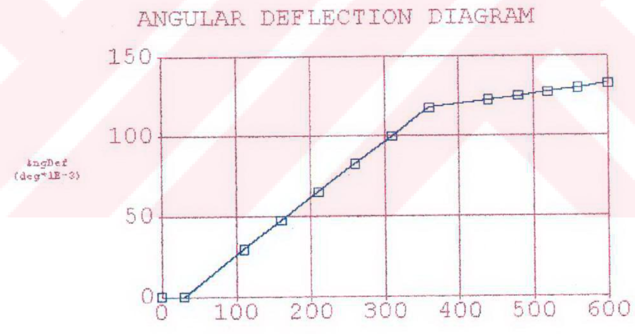


Figure 7-4 Torque and Angular Deflection Diagrams

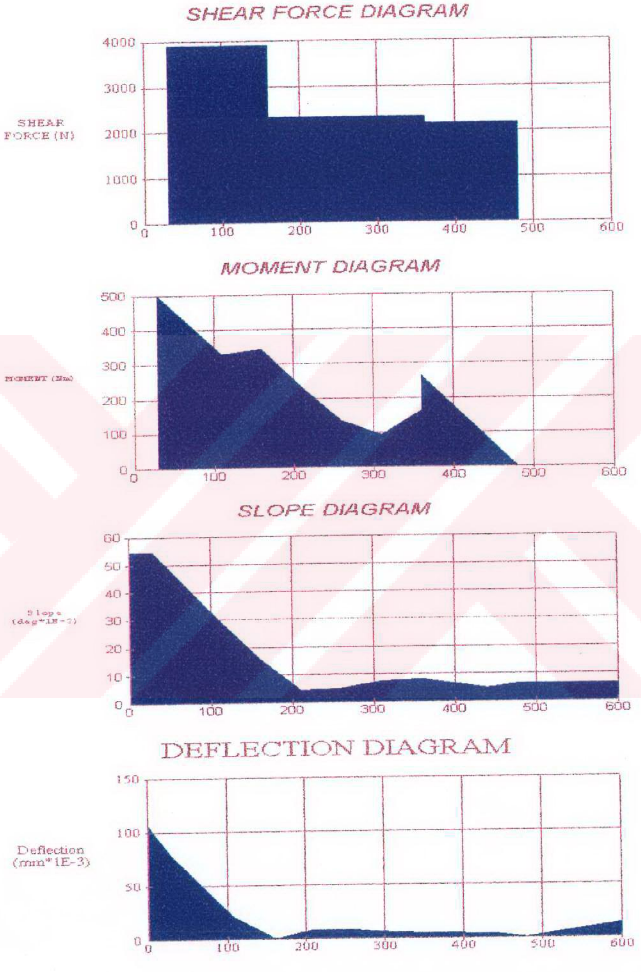


Figure 7-5 Maximum Loading Diagrams

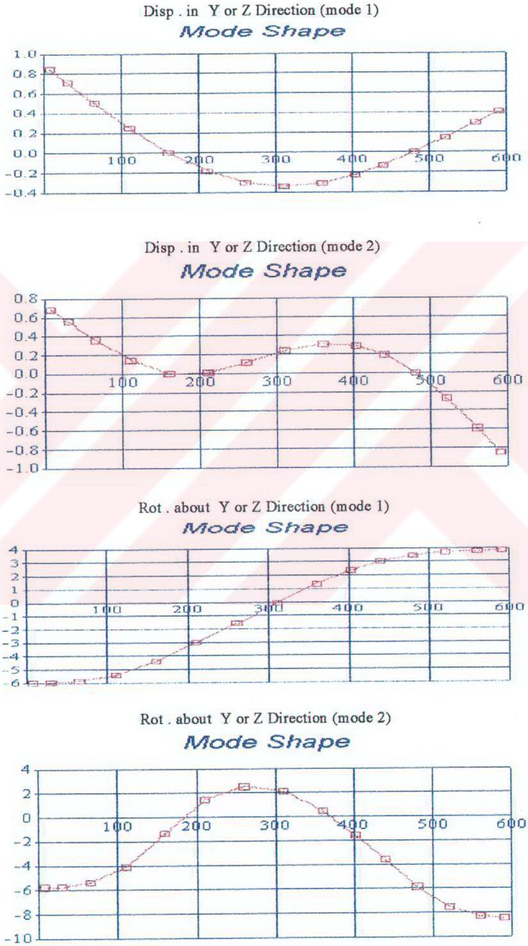
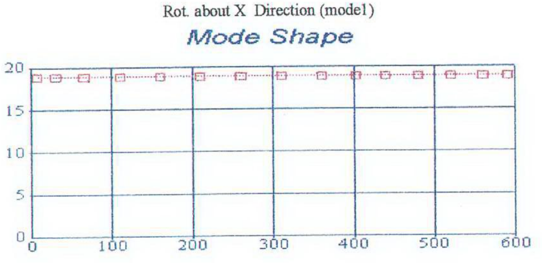


Figure 7-6 First Two Mode Shapes of the Flexural System



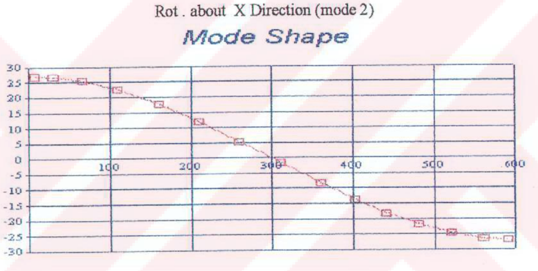


Figure 7-7 First Two Mode Shapes of the Torsional System

Table 7-5 First Two Critical Speeds of Flexural and Torsional System

Mode#	Critical Speeds(rpm)	
	Flexural	Torsional
1	37618	0
2	72095	161954

Table 7-6 Combined Numerical Results of the Analysis

ST #	Diameter mm	MAX SHEAR Nm	MAX BENDING Nm	MAX SLOPE deg E-3	MAX DEF mm E-3	MAX ANG DEF deg E-3	STRENGTH DISTRIBUTION
1	50	0	0	54.486	105.3648	0	1000
2	50	0	0	54.486	77.3792	0	1000
3	50	3905.125	328.0244	29.8459	19.9879	29.5166	3.6751
4	50.4	3905.125	343.1108	15.8073	0	47.3847	6.0519
5	50.4	2333.362	232.2793	4.235	8.1815	65.2586	5.046
6	50.8	2333.362	131.9287	4.7532	8.2616	82.5707	12.107
7	50.8	2333.362	89.8867	7.4138	5.1146	99.8828	10.4853
8	50.4	2333.362	161.1737	8.1267	4.9072	117.7567	10.4476
9	50.4	2184.652	87.3861	4.7634	4.0048	122.5204	14.8324
10	50	2184.652	0	6.2368	0	124.9796	158.5985
11	50	0	0	6.2368	4.3518	127.4389	158.5985
12	49.6	0	0	6.2368	8.7036	129.9783	155.1124
13	49.6	0	0	6.2368	13.0553	132.5178	155.1124
$VOLUME(mm^3) = 1188579$							

From the above table, it can be examined that critical shaft sections for rigidity and strength which are shown shaded, are at different stations and 50 mm shaft diameter is a good assumption for strength. On the other hand, critical speed of shaft is far from the operating shaft speed.

CASE 1-B : The same problem is examined under "Conventional Design" process. Design constraints for rigidity are not taken in to consideration by entering higher numbers. Table 7-7 shows constraints of the problem. Convergence of strength distribution to the factor of safety is examined.

Table 7-7 Design Constraints for Conventional Design Process

Factor of Safety	3
Shaft Speed	1500 rpm
Maximum Allowable Slope	1 deg
Maximum Allowable Deflection	1 mm
Maximum Allowable Ang. Deflection	1 deg

The conventional design process for the power transmission shaft design problem is to select some initial values for the design variables. Initial guess of variables are accomplished by design procedure where variables are assumed dependent to each other with fillet radii. There is no objective function. Main purpose is an acceptable design.

Table 7-8 Combined Numerical Results of the Conventional Design Process

ST #	Diameter mm	MAX SLOPE deg E-3	MAX DEF. mm E-3	MAX ANG DEF deg E-3	STRENGTH DISTRIBUTION
1	46.71875	71.3765	137.9458	0	1000
2	46.71875	71.3765	101.2855	0	1000
3	46.71875	39.0499	26.1444	38.7229	3.0098
4	47.11875	20.67	0	62.1172	4.9708
5	47.11875	5.5208	10.6892	85.5115	4.1385
6	47.51875	6.2232	10.7842	108.1261	9.9476
7	47.51875	9.6985	6.6758	130.735	8.6067
8	47.11875	10.6283	6.4363	154.1293	8.5729
9	47.11875	6.2405	5.2502	160.3662	12.1694
10	46.71875	8.1752	0	163.5936	131.272
11	46.71875	8.1752	5.7055	166.821	131.272
12	46.31875	8.1752	11.411	170.1631	128.1396
13	46.31875	8.1752	17.1165	173.4994	128.1396

$VOLUME(mm^3) = 1038382$

Table 7-8 shows that, convergence obtained to an acceptable design for strength by 0.03% accuracy.

CASE 1-C : The same problem is examined under "Optimum Design" process. Table 7-9 shows constraints for this case.

Table 7-9 Design Constraints for Optimum Design Process

Factor of Safety :	3
Shaft Speed :	1500 rpm
Maximum Allowable Slope :	1 deg
Maximum Allowable Deflection :	1 mm
Maximum Allowable Ang. Deflection :	1 deg
Maximum Allowable Diameter :	100mm
Minimum Allowable Diameter :	5 mm
Maximum Number of Stages	5
Vector of Initial Step Size	0.5

The optimum design process takes into account all the constraints simultaneously, and iteratively improves the design, while minimizing the "Volume".

Table 7-10 Combined Numerical Results of the Optimization Process

ST #	Diameter mm	MAX SLOPE deg E-3	MAX DEF mm E-3	MAX ANG DEF deg E-3	STRENGTH DISTRIBUTION
1	46.6966	69.0475	131.488	0	1000
2	46.6966	69.0475	96.1034	0	1000
3	46.6966	36.8533	24.2661	38.7975	2.9994
4	47.1167	18.5066	0	62.1975	4.9702
5	47.1167	3.195	8.6877	85.5917	3.3671
6	52.448	4.8486	7.9991	100.8287	13.2994
7	52.448	7.249	4.4185	116.0656	10.953
8	51.204	7.8186	4.4795	132.8389	10.9448
9	51.204	4.18	4.3091	137.3102	12.1098
10	42.026	7.1955	0	142.2401	97.2551
11	42.026	7.1955	5.0215	147.1701	97.2551
12	15.58	7.1955	10.043	407.8204	5.1859
13	15.58	7.1955	15.0645	668.4764	5.1859

$VOLUME(mm^3) = 972227$

This case have been tried for an initial step size at 0.1 for the first time but it failed. This can sometimes happen in optimization. In such cases the user can slightly change the stepsize. After changing the step size to 0.5 the results shown in Table 7-10 is obtained. The design is in acceptable limits by %0.02 accuracy but this time volume decreased by 6.37 % with respect to design procedure. If the maximum number of stages is increased, more accurate results can be obtained. Hence, it should increase the process time.

CASE 1-D : The same problem is examined for hollow shaft under "Optimum Design" process. Table 7-11 shows constraints for this case.

Table 7-11 Design Constraints for Hollow Shaft Under Optimum Design Process

Factor of Safety :	3
Shaft Speed :	1500 rpm
Maximum Allowable Slope :	1 deg
Maximum Allowable Deflection :	1 mm
Maximum Allowable Ang. Deflection :	1 deg
Maximum Allowable Wall-Tickness :	5 mm
Maximum Allowable Diameter :	100mm

Table 7-12 Combined Numerical Results of the Optimization Process for Hollow Shaft

ST #	Diameter mm	Inside Diameter mm	MAX SLOPE deg E-3	MAX DEF mm E-3	MAX ANG DEF deg E-3	STRENGTH DISTRIBUTION
1	52.43721	35.16395	50.4049	92.7112	0	1000
2	52.43721	35.16395	50.4049	66.9067	0	1000
3	52.43721	35.16395	24.5651	16.2935	30.5885	3.0007
4	54.94423	35.16395	12.6292	0	45.7854	6.4822
5	54.94423	35.16395	2.7391	6.208	60.9879	4.8279
6	57.62336	35.16395	3.6143	5.9626	73.1293	15.1089
7	57.62336	35.16395	5.5045	3.507	85.2764	12.6659
8	56.68976	35.16395	5.9953	3.6464	98.3752	12.5738
9	56.68976	35.16395	3.4294	3.2106	101.872	14.1586
10	51.02855	35.16395	5.1988	0	104.8013	132.3354
11	51.02855	35.16395	5.1988	3.6297	107.7248	132.3354
12	45.16055	35.16395	5.1988	7.2596	113.572	77.062
13	45.16055	35.16395	5.1988	10.8894	119.4134	77.062

$VOLUME(mm^3) = 772242$

CASE 1-E : The same problem is examined for hollow shaft under "Optimum Design" process again. Hence, this time constraint for maximum allowable deflection decreased to 0.05 mm. Table 7-13 shows constraints for this case.

Table 7-13 Design Constraints for Hollow Shaft Under Optimum Design Process Constrained by Maximum Allowable Deflection

Factor of Safety :	3
Shaft Speed :	1500 rpm
Maximum Allowable Slope :	1 deg
Maximum Allowable Deflection :	0.05 mm
Maximum Allowable Ang. Deflection :	1 deg
Maximum Allowable Wall-Tickness :	5 mm
Maximum Allowable Diameter :	100mm

The important factor affecting the selection of that value is the maximum deflection that is obtained from the previous case (0.092 mm). In order to control the program for rigidity, less maximum allowable deflection is entered.

Table 7-14 Combined Numerical Results of the Optimization Process for Hollow Shaft Constrained for Deflection

ST #	Diameter mm	Inside Diameter mm	MAX SLOPE deg E-3	MAX DEF mm E-3	MAX ANG DEF deg E-3	STRENGTH DISTRIBUTION
1	54.7119	36.479	32.174	49.8228	0	1000
2	54.7119	36.479	32.174	33.4083	0	1000
3	54.7119	36.479	9.4075	6.1554	25.6586	2.9998
4	67.4181	36.479	4.6272	0	31.7637	12.9497
5	67.4181	36.479	0.6243	2.0799	37.8688	8.6445
6	77.5562	36.479	1.1677	1.8128	41.2223	39.9744
7	77.5562	36.479	1.7029	0.9105	44.5701	32.3728
8	75.2213	36.479	1.8257	1.0659	48.3822	31.997
9	75.2213	36.479	0.8919	1.1655	49.4025	33.9487
10	56.0129	36.479	2.0696	0	51.3057	180.6868
11	56.0129	36.479	2.0696	1.4443	53.2032	180.6868
12	46.486	36.479	2.0696	2.8886	58.5057	82.2867
13	46.486	36.479	2.0696	4.3328	63.814	82.2867

$VOLUME(mm^3) = 1370916$

The numerical results that is presented in Table 7-14 shows that the design is again in acceptable limits. This time in order to achieve the maximum allowable deflection optimization process increased outside diameter for some of the stations.

Another point that can be added for the hollow shaft cases by examining the stations 12 and 13, the minimum wall-thickness constraint works at these sections.

CASE 2: Some alternating loads are added to the sample problem. Types and intensity of the loads are shown in Figure 7-8 and static versus dynamic responses are compared in X-Y plane.

Table 7-15 Included Alternating Loads with its Station Numbers and Distances

ST #	DIST DX (mm)	NOTCH	NOTCH RAD (mm)	X-Y plane FORCE (alternating) (N)	X-Y plane MOMENT (alternating) (Nm)	Alternating Load Type
1	0	Smooth	0	0	0	
2	30	Smooth	0	1500	250	$\cos(\omega_1 t)$
3	80	Incr. Dia.	0.2	0	0	
4	50	Smooth	0	0	0	
5	50	Incr. Dia.	0.2	0	0	
6	50	Smooth	0	0	0	
7	50	Decr. Dia.	-0.2	0	0	
8	50	Keyway	0	1000	100	$\cos(\omega_2 t)$
9	80	Decr. Dia.	-0.2	0	0	
10	40	Smooth	0	0	0	
11	40	Decr. Dia.	-0.2	0	0	
12	40	Smooth	0	0	0	
13	40	Keyway	0	0	0	

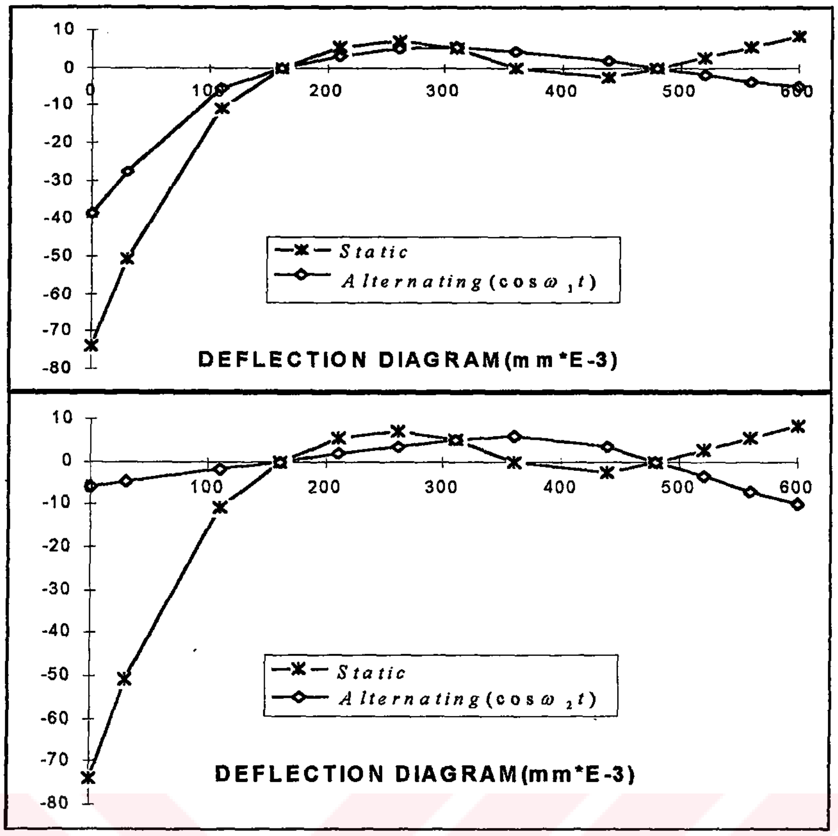


Figure 7-8 Static Versus Dynamic Deflections at X-Y Plane

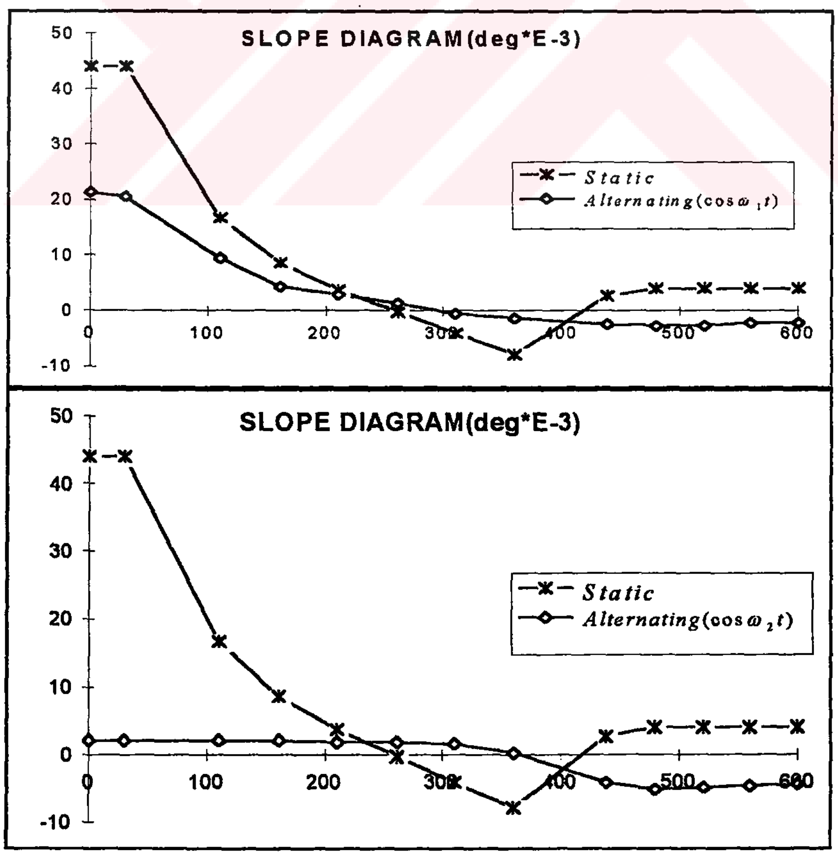


Figure 7-9 Static Versus Dynamic Slope at X-Y Plane

Table 7-16 Combined Numerical Results of Dynamic Loading

ST #	Diameter mm	MAX SHEAR Nm	MAX BENDING Nm	MAX SLOPE deg E-3	MAX DEF mm E-3	MAX ANG DEF deg E-3	STRENGTH DISTRIBUTION
1	50	0	0	61.4717	125.3131	0	1000
2	50	0	0	61.1196	94.0964	0	1000
3	50	5147.815	352.2783	37.5104	27.0865	29.5166	3.4526
4	50.4	5147.815	331.4363	24.3892	0	47.3847	6.2198
5	50.4	2355.699	224.4887	13.154	15.8859	65.2586	5.1901
6	50.8	2355.699	136.7001	7.5	23.1381	82.5707	11.8513
7	50.8	2414.657	170.6337	8.0034	23.6221	99.8828	6.6179
8	50.4	2414.657	196.24	12.6436	17.5541	117.7567	9.1646
9	50.4	1794.768	71.7907	12.309	9.4124	122.5204	17.7295
10	50	1794.768	0	14.0243	0	124.9796	158.5985
11	50	0	0	14.2182	9.8168	127.4389	158.5985
12	49.6	0	0	14.7208	19.9181	129.9783	155.1124
13	49.6	0	0	14.9214	29.8448	132.5178	155.1124

In this example, for conservative design, different types of dynamic displacements are calculated separately and their absolute values are used in finding their combination. That is why numerical values of maximum deflection and maximum slopes are increased without changing the critical sections that are shown shaded in Table 7-16.

CHAPTER 8

CONCLUSIONS AND FUTURE RECOMMENDATIONS

8.1 CONCLUSION

Development and increased usage of software packages have greatly improved efficiency of engineering design. Due to such programs, design, analysis and documentation are made easier and more reliable for the designer.

In this study a general program for the design of power transmission shafts is developed. With the developed software a solid or stepped shaft with different cross sections can be designed. The reaction forces at the supports are calculated by the program. The loading diagrams, together with shear force, bending moment, deflection and slope diagrams can be obtained in two dimensional drawings.

Also in this study dynamic behavior of shafts for lateral and torsional systems is determined. While finding dynamic behavior of the shaft, alternating forces, unbalances and flexible bearings are taken into consideration, not only in finding critical speeds but also in finding dynamic responses. These are generated as new design constraints for conservative design purposes.

Two kinds of processes which are "Conventional Design" and "Optimum Design", are included to increase the capability of the software.

In conventional design process, initial guess of variables are done by design procedure where variables are assumed dependent to each other with fillet radii. This process only takes care of the design constraints like strength, rigidity, and stability. If one of the design variables is exceeded, then design variables are adjusted and process is repeated until the variables are in the allowable limits.

The developed conventional design process can play role as a design assistant and following benefits can be obtained.

- Decrease the process time as variables are assumed dependent

- Finds the minimum diameter that satisfies the design constraints

- A clever guess in determining the initial values of the variables for optimization process

In contrast optimum design process takes into account all the variables independently and improves the design, while minimizing the volume that result in better design. It is more formal then the conventional design process. However, it increases the process time depending on the number of sections and also it does not guarantee a solution.

Implicit and explicit constraints are determined and implemented to the Constrained Rosenbrock Algorithm for the optimal design of power transmission shafts. The developed optimum design process increased the capability of the program and following benefits are obtained.

- Finds minimum diameter at each section that satisfies the design constraints

- Another design alternative to try

- A new design procedure which is more accurate

- It minimizes the volume

The program is helpful to the design engineer who wishes to design a power transmission shaft, since it decreases the calculation time and allows the designer to try different design alternatives in a short time. The errors made during the manual design process, like inaccurate calculations, incorrect data read from tables or charts are also eliminated through the use of the program. Two dimensional drawings can be viewed, entering the inputs and presenting the outputs for initial check. Tables and charts that presents the numerical and graphical results of the calculation can be copied to an other Windows application in order to compare the results.

8.2 FUTURE RECOMMENDATIONS

Forced responses of power transmission shafts were calculated for harmonic excitations. In this study alternating forces repeating every revolution is set to harmonic motion without knowing the phase differences. They are taken into account for conservative design. If dynamic behavior of other machine element can be included, more accurate results can be obtained.

Optimization process can be change individual to individual only in determining the implicit variables that constraints the shaft for physically meaningful solution. This process can be improved by enlarging the implicit variables, including distance between the stations other then the diameters at the sections and the inside diameter.

The data needed for manufacturing of the shaft could be added to its database engine. Writing another subroutine that gives the manufacturing information about the calculated results can be very useful.

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APPENDIX A

USER'S GUIDE

This appendix has been designed to provide the user an overview of the developed "Shaft Design" program.

A.1. SYSTEM REQUIREMENTS

Minimum IBM compatible 486 based computer, VGA graphics driver (640×480,256 colors), 640 Kb base memory , MS-DOS 6.0 , MS-WINDOWS 3.1, MS-WINDOWS 95 (previous versions are not tested).

The program consists of five subdirectories under the directory "SHAFT". Main calculation of the program is developed by using TC 3.0 in MS-DOS environment consist of Design Sub-Programs (*.h) and a main program (shaft.cpp) file. User interface part of the program is developed by using Visual Basic 4.0 in MS-Windows environment. The sub-programs are located in the directory structure shown in Figure A.1.

After running the program, "PASSWORD" menu is placed on the screen. If the password is written right in order, "MAIN" menu takes its place as shown in Figure A.2. Otherwise there is only three chances to enter the right password.

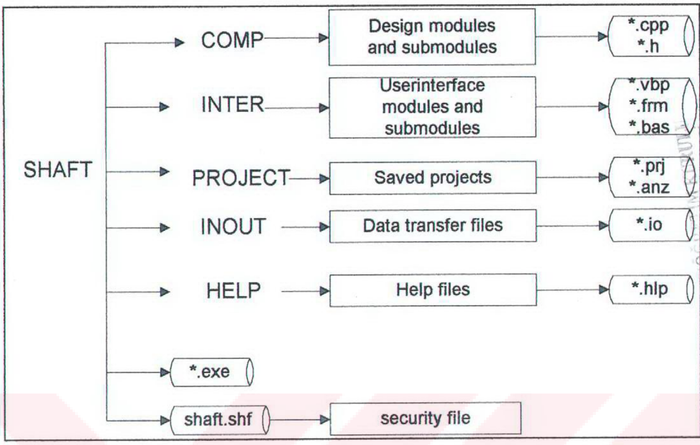


Figure A.1. Directory Structure of the Sub-programs

Now running an application is considered and described as an example by showing the windows designed for the developed program.

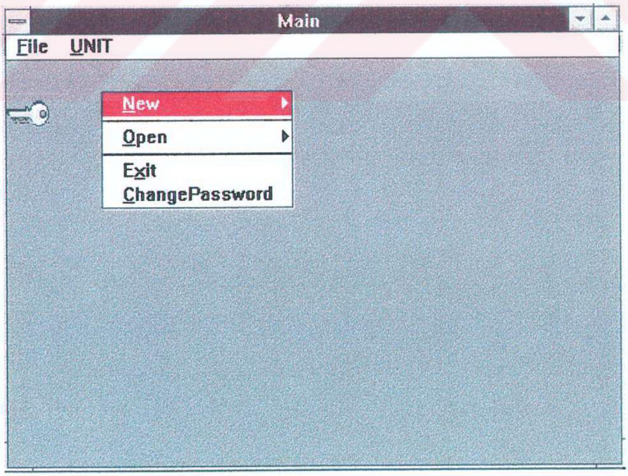


Figure A.2 Main Menu

Main Menu :

1. Click on the "Unit" menu to change the unit system.

2. Click on the "KEY" icon. This icon will move and open the selections under the "File" menu.

3. Click on one of the selections as shown in Figure A.2.

The user can open a previous project or create a new project to analyze or design a shaft system. If the user wants to open a previous project, a dialog box that shows the names of the files appears as shown in Figure A.3.

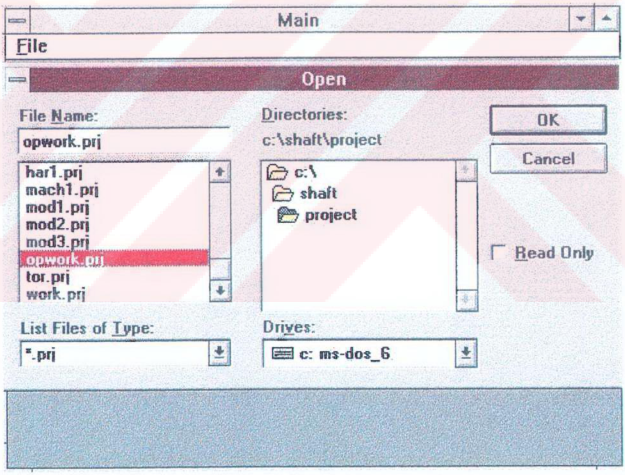


Figure A.3 Opening a Saved Project

After choosing a specified file, the stored design conditions are placed to their field and first design window appears as shown in Figure A.4.

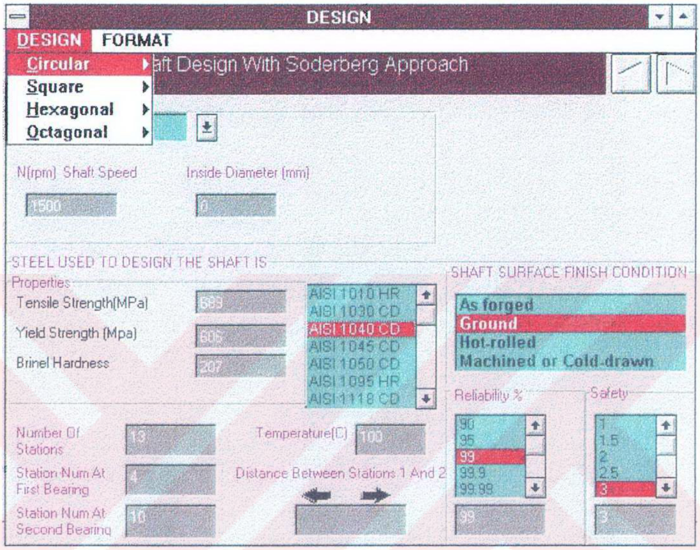


Figure A.4 First Design Window

First Design Window :

1. Choose one of the shaft types by pressing the "Design" menu.

2. Click on the "Design Theories" and choose one of them.

3. Enter the shaft speed using keyboard.

4. Click on one of the selections in the box where "AISI" steels are specified or enter a numerical value in the "Tensile Strength", "Yield Strength", "Brinell Hardness" boxes.

5. Click on one of the selections in the "Shaft Surface Finish Condition" box.

6. Select the reliability and factor of safety of the shaft from the appropriate boxes.

7. Enter the environmental temperature in the "Temperature" box.

8. Enter the number of stations, the station number at first bearing and the station number at second bearing

9. Enter the distance between stations and press "Return" from the keyboard until the box does not give a response. In order to control the values click on the arrows over the box.

10. Click on the arrows which are at the right corner of the first design window to open the "Bearing Window" as shown in Figure A.5. or to return back to the "Main Menu".

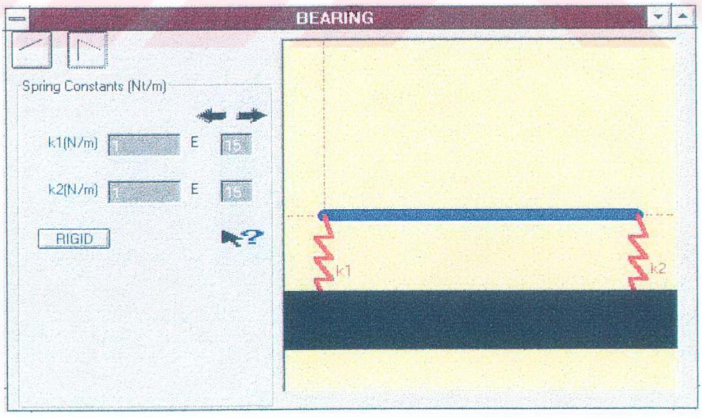


Figure A.5 Bearing Window

Bearing Window :

1. If the bearings are rigid then press the "Rigid" button in the bearing window.

2. If the bearings are not rigid then enter the spring constants.

3. By pressing the right arrow load the second design window.

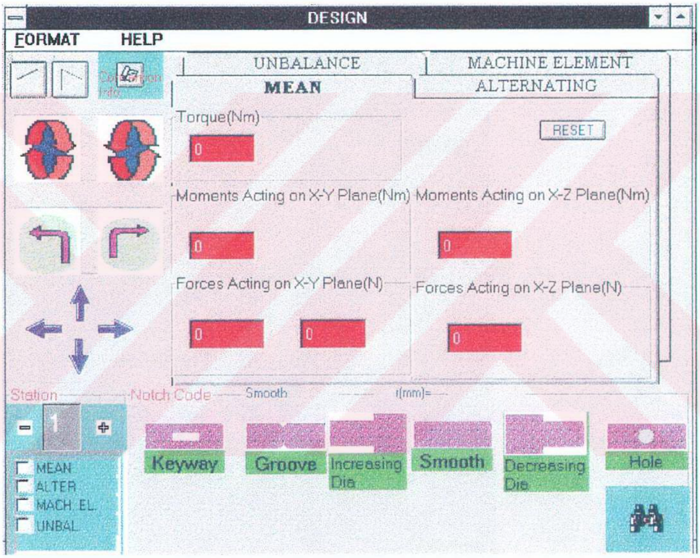


Figure A.6 Second Design Window

Second Design Window :

1. Write the numerical values of the mean loads (if there is any) in the boxes under "MEAN" tab for the station number shown under the "Station" label. The same thing can be done for

alternating loads, machine elements and unbalances by pressing the "Alternating", "Machine Element" and "Unbalances" tabs.

2. Pressing the icons which are near the loads can be helpful in the sign conventions of the mean loads.

3. If the values at a station are entered wrong, pressing the "RESET" button will clear them.

4. Define the notch code by clicking on one of the icons that are shown in the "Notch Code" frame.

5. Increase or decrease the station number by clicking on the "+" or "-" icon under the "Station" label.

6. In order to see the sign convention that is used while preparing the developed program, click on the "Convention Info" icon.

7. Click on the "View" icon in order to examine the drawing of the shaft as shown in Figure A.7.

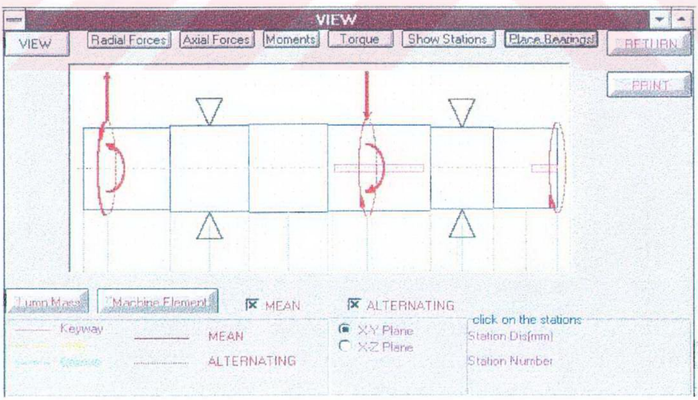


Figure A.7 View Window while Showing the Input Values

View Window :

1. Click on the "X-Y plane" or "X-Z plane" box to select the plane.

2. Sign on the "Alternating" and "Mean" boxes or both of them.

3. Only the schematic representation of the shaft appears by pressing on the "View" button.

4. In order to view radial forces, axial forces, torques, moments, stations and machine elements, press the appropriate buttons.

5. Press the "Return" button in order to return to the second design window.

In the second design window by pressing the "Convention Info" icon, user can examine the sign convention as shown in the Figure A.8.

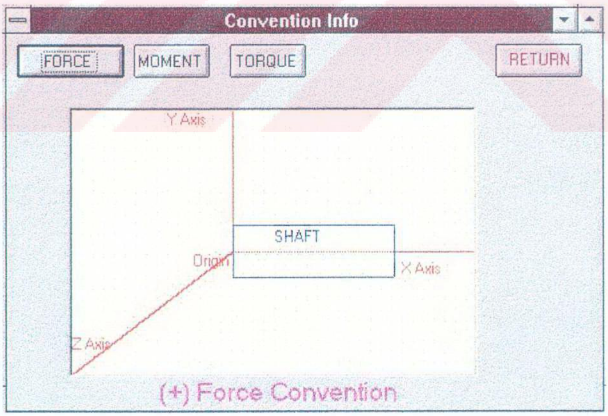


Figure A.8 Convention Window

Convention Window :

1. In order to examine the sign conventions that are used in the calculation, press the "Force", the "Moment" and the "Torque" buttons. Order is not important.

2. By pressing the "Return" button, return to the second design window.

After returning back to the second design window by clicking on the right arrow icon the user can activate the constraint window as shown in the Figure A.9.

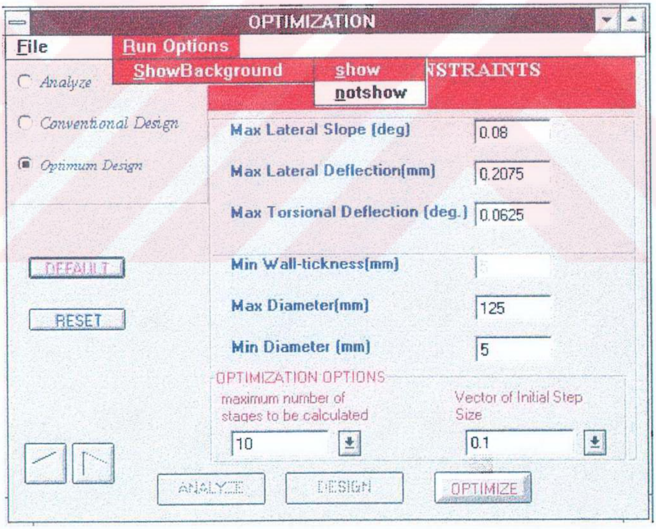


Figure A.9 Constraint Window

Constraint Window :

1. Sign one of the "Analyze", "Conventional Design" or "Optimum Design" selections. This will activate the related constraints.
2. Enter the constraints which are activated under the "Design Constraints" label or press the "Default" button in order to enter the default values.
3. Click on the "Save As" selection under the "File" menu in order to save the application as shown in the Figure A.10.
4. Press the button which is activated by the selection at the bottom of the window in order to run the calculation program under MS-DOS environment.
5. Click on the right arrow icon in order to examine the results.

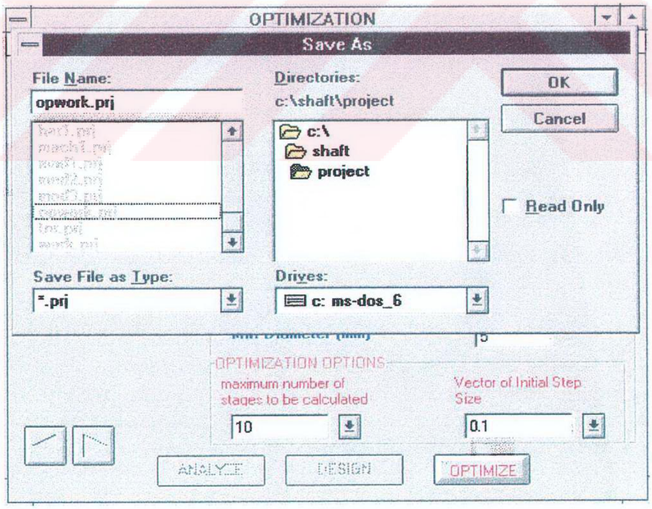


Figure A.10 Saving the Application

First Result Window:

1. Select one of the planes in which the results will be examined which are the "X-Y plane", "X-Z plane" and "Maximum".
2. Sign the other selections that are wanted to be examined and click on the "Graph" icon in order to see the graphical results or click on the "Table" icon in order to see the numerical results.
3. For the specified selections if the "Table" icon is clicked after clicking on the "Graph" icon, both of the results can be observed as shown in Figure A.11.
4. Press the "View" icon in order to examine the calculated diameters at the stations as shown in Figure A.12.

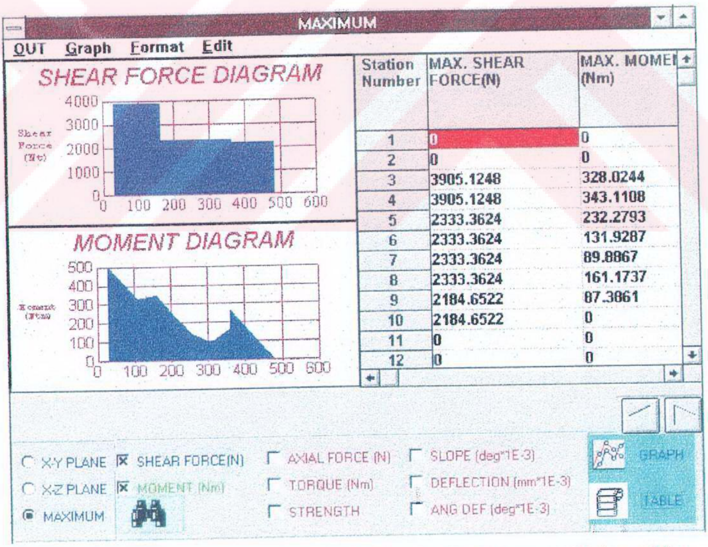


Figure A.11 First Result Window while Showing the Maximum Values of the Selection

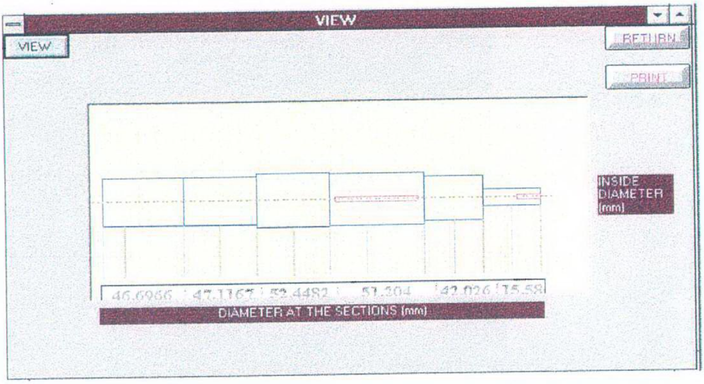


Figure A.12 View Window while Showing the Results of the Optimum Design

View Window :

1. Press the "View" button in order to view the picture as shown in Figure A.12.

2. Press the "Return" button in order to return back to the first result window.

After returning back to the first result window, byclicking on the right arrow icon the user can activate the compare window as shown in the Figure A.13. This window is created in order to show the accuracy of the calculation by comparing the calculated values of the constraints with the allowable limits that are specified in the design windows. If the accuracy is satisfactory, the user can continue examining the other results as shown in Figure A.14 by clicking on the right arrow.

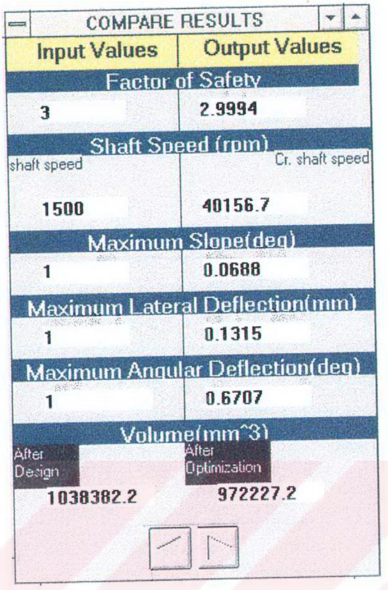


Figure A.13 Compare Window

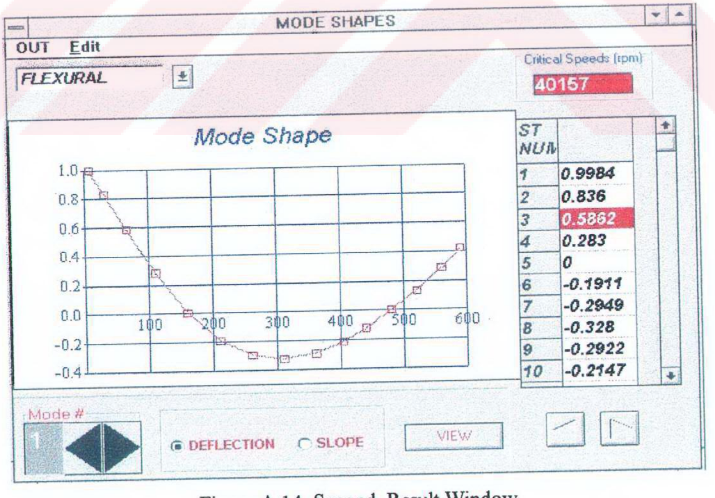


Figure A.14 Second Result Window

Second Result Window:

1. In order to examine the numerical results of the mode shape and the critical speed for the first mode, click on the "Table" selection under the "Out" menu.

2. Increase or decrease the mode number by pressing the right arrow or the left arrow under "Mode" and examine the results for the other modes.

3. Click on the "Graph" selection under the "Out" menu in order to make the "System" box visible and choose one of the systems which are "Flexural" and "Torsional".

4. Sign the "Deflection" or the "Slope" box and press the "View" button in order to examine the numerical results together with the graphical results as shown in the Figure A.14.

5. In order to examine the results for other modes press the left arrow or the right arrow under the "Mode" frame.

A.2. OTHER FUNCTIONS

File Menu :

New : Create a new project

Open : Open a previous project

Save : Saves the opened project without changing the file name

Save As : Saves the opened project by changing the file name

Delete : Deletes the opened project

Change Password : This button is used for changing the password

Unit Menu :

SI : Converts the input values to "SI" unit system

British : Converts the input values to "British" unit system

Format Menu :

Color : Changes the color of the form

Font : Changes the font of the text

Print : Prints the form

Run Options Menu:

Show Background : Shows the flow of the program at the background

Edit Menu :

Copy : Copies the selected graph or selected table

Graph Menu :

Min Max : Shows the minimum and maximum values

StdDev : Calculates the standard deviation of the selected curve and draws on the selected graph

BestFit : Draws the line that fits the selected curve

Mean : Shows the mean value of the selected graph

View Icons or Buttons :

Draws 2D graph of the input values or the results to be visualized by the user

Convention Info Icon :

Draws 2D graph of the force, torque and moment conventions

APPENDIX B

SAMPLE PROBLEMS

This appendix has been designed to check some of the critical calculations that are implemented to the developed program. Some of the problems that are solved in this section are taken from the literature and adapted to solve with the program. Comparing the results with the answers given in the literature will give a brief idea about these critical calculations.

It must be remembered that the simplicity of these problems that are computed, does not show the capability of the developed program.

Sample Problem 1 :

Calculation for strength is tested under "Optimization Process" by entering the parameters given in Table B-1 (Shigley [3], Example 18-2). Although original problem taken from Shigley [3] concerns diameter calculation at the most critical section of a stepped shaft, where stress concentration factors are given, the developed program finds these factors according to the fillet radii. Assuming the shaft as if it is uniform, the diameter at the most critical section can be found without taking the stress concentration factors in to account. Other parameters that are not given in the example are assumed as well.

Table B.1 Input Information Given in Shigley [3] for Sample Problem 1

PARAMETES	NUMERICAL VALUES	
Design Theory	Modified Goodman	
Number of station =	6	
Station number at first support=	2	
Station number at second support=	4	
Force at station 3 =	-2000	lb.
Force at station 5 =	-1100	lb.
Torque at station 3 =	3300	lb.in
Torque at station 5 =	-3300	lb.in
Bearing type	Rigid	
Factor of Safety	1.8	
Tensile Strength	80	kpsi
Yield Strength	66	kpsi
Distance Between Stations 1 and 2	0.5	in.
Distance Between Stations 2 and 3	3	in.
Distance Between Stations 3 and 4	3	in.
Distance Between Stations 4 and 5	1.75	in.
Distance Between Stations 5 and 6	1	in.

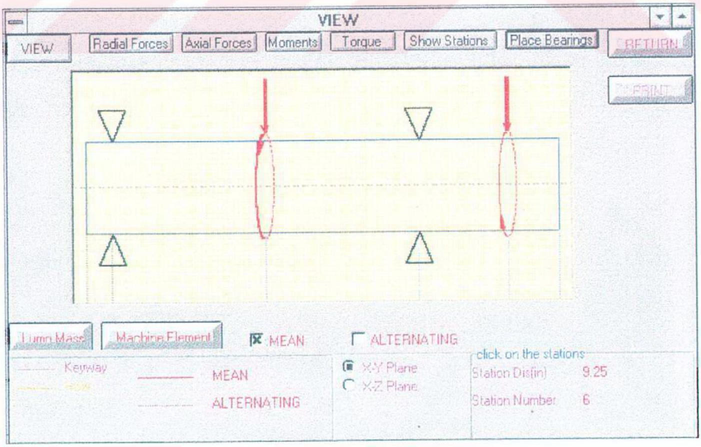


Figure B.1 Loading Diagram of Sample Problem 1

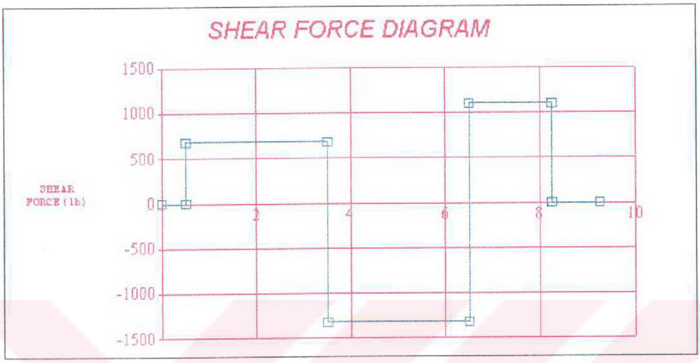


Figure B.2 Shear Force Calculated by the Developed Program

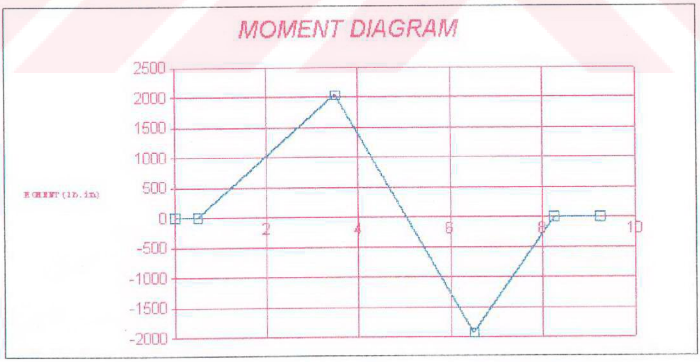


Figure B.3 Moment Diagram Calculated by the Developed Program

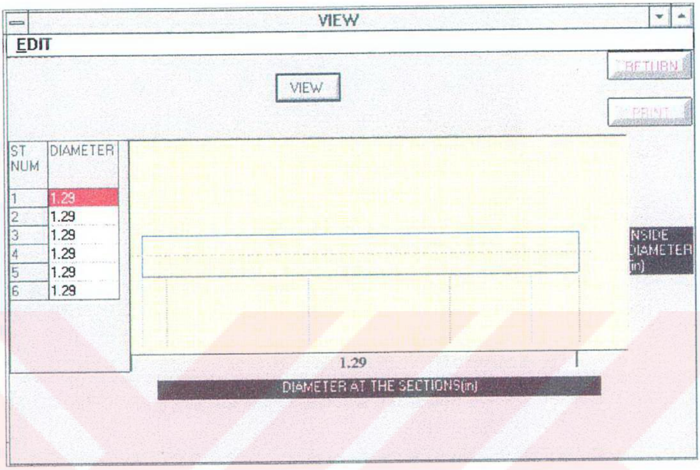


Figure B.4 Diameter Calculated by the Developed Program

Although diameter that is found by Shigley [3] was 1.26 in. , the result that is calculated by the developed program is reasonable as the problem is adapted for the unknown parameters.

Sample Problem 2 :

Simply supported stepped shaft is tested for deflection under "Analyze Process" by entering the parameters given in Shigley [3] (Section 3-6 "Finding Deflections by Numerical Integration"). The parameters which are used in the developed program, are given in Table B.2.

Table B.2 Input Information Given in Shigley [3] for Sample Problem 2

PARAMETES	NUMERICAL VALUES	
Number of station =	9	
Station number at first support=	2	
Station number at second support=	8	
Force at station 4 =	-600	lb.
Force at station 6 =	-1000	lb.
Bearing type	Rigid	
Distance Between Stations 1 and 2	0.375	in.
Distance Between Stations 2 and 3	0.375	in.
Distance Between Stations 3 and 4	1.625	in.
Distance Between Stations 4 and 5	7	in.
Distance Between Stations 5 and 6	5	in.
Distance Between Stations 6 and 7	1.625	in.
Distance Between Stations 7 and 8	0.375	in
Distance Between Stations 8 and 9	0.375	in.

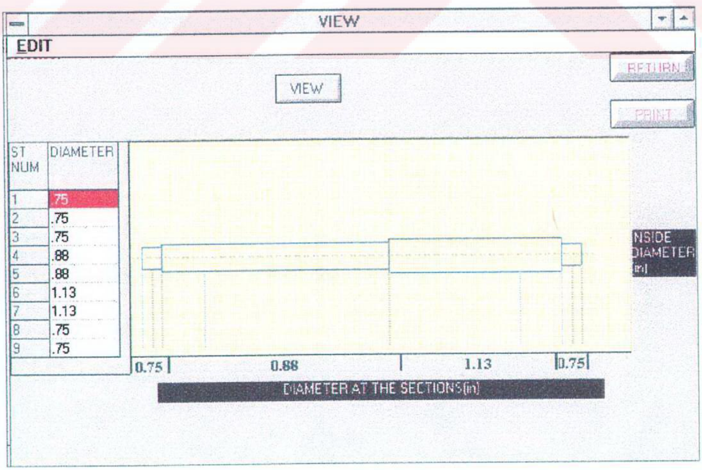


Figure B.6 Simply Supported Stepped Shaft

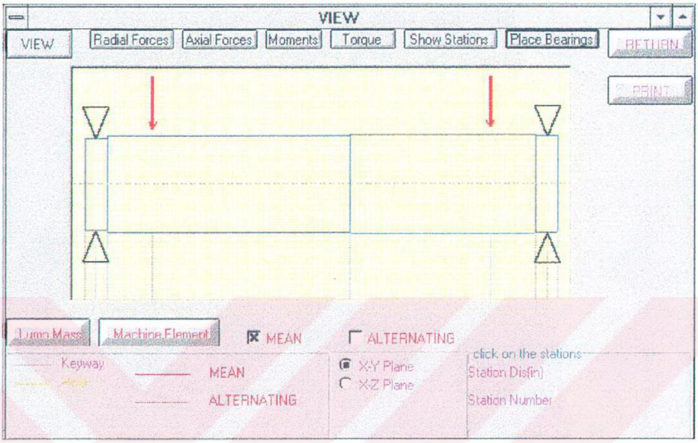


Figure B.7 Loading Diagram of Sample Problem 2

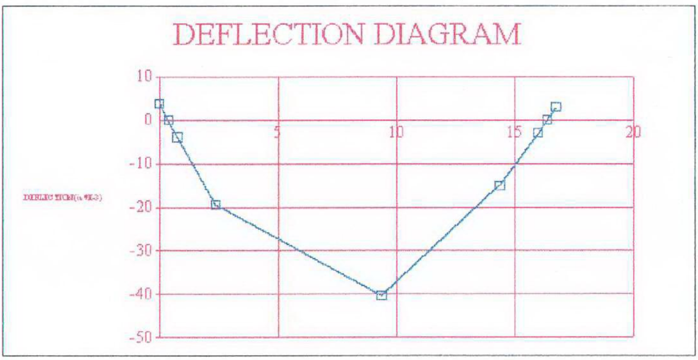


Figure B.8 Deflection Diagram Calculated by the Developed Program

Table B.3 Numerical Results of the Sample Problem 2

STATION #	DEFLECTIONS(in E-3)	
	Dev. Program	Shigley [3]
2	0	0
3	-3.8449	-3.844
4	-19.4881	-19.5
5	-40.4145	-40.408
6	-15.0857	-15.084
7	-2.9491	-2.9488
8	0	0

As the results that are given in Shigley [3] are calculated with the same numerical method used in the developed program, they are very close to each other and none of them introduces the exact values. On the other hand accuracy between numerical values shows that method is implemented to the program correct.

Sample Problem 3

Eigensolver of the flexural system implemented to the developed program is tested under a simple problem where its parameters are given in Table B.4. Results of the analyze can be seen in Figure B.10-B.12. First three critical speeds are compared with the exact values calculated by the formulations given in Kelly [29] (Table 9.4, "Natural Frequencies and Mode Shapes for Beams").

Table B.4 Input Information for Sample Problem 3

PARAMETES	NUMERICAL VALUES
Number of station =	9
Station number at first support=	1
Station number at second support=	9
Bearing type	Rigid
Distance Between Stations	50 mm.
Shaft Diamter	30 mm.

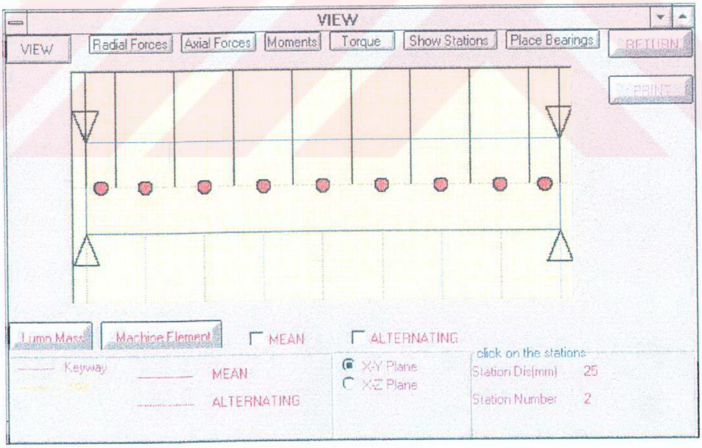


Figure B.9 Lumped Mass Model of Sample Problem 3

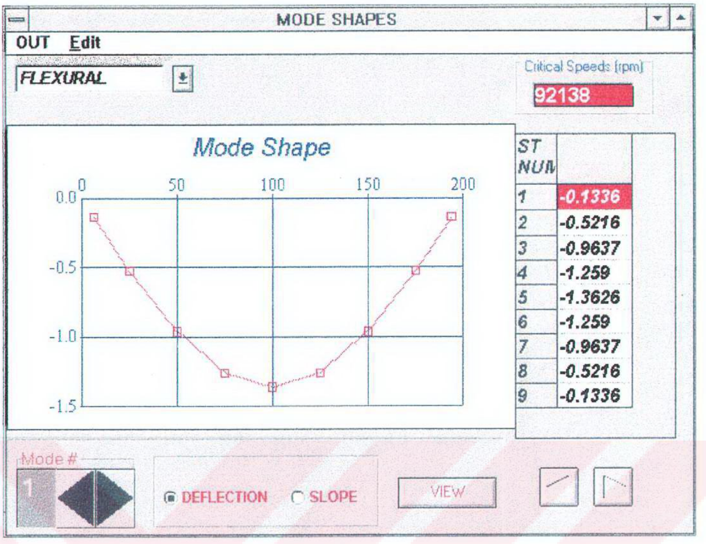


Figure B.10 First Mode Shape and Critical Speed Calculated by the Developed Program

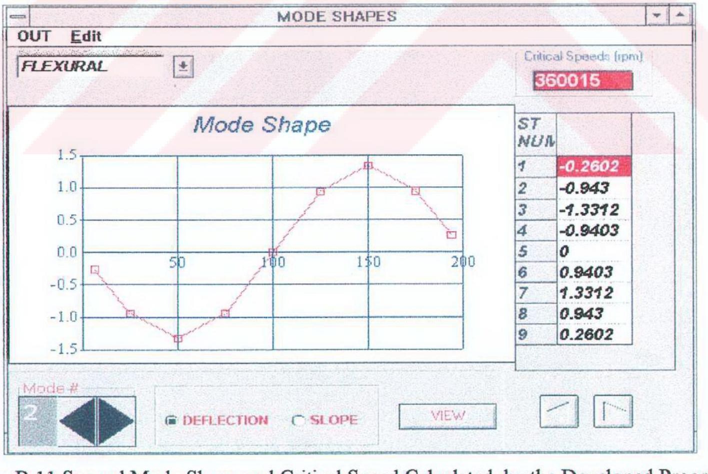


Figure B.11 Second Mode Shape and Critical Speed Calculated by the Developed Program

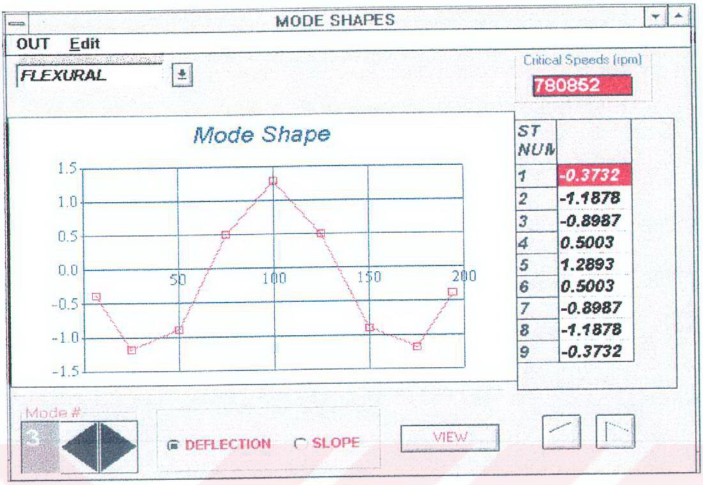


Figure B.12 Third Mode Shape and Critical Speed Calculated by the Developed Program

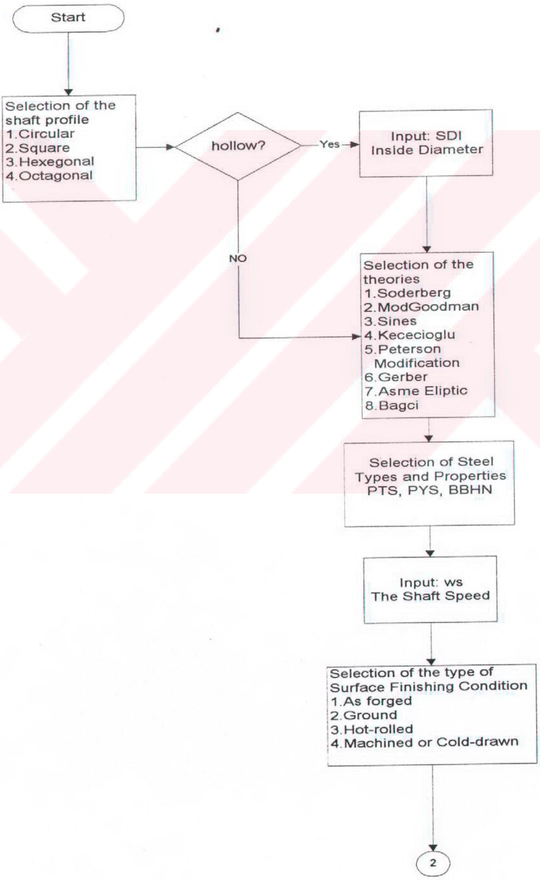
Table B.5 Numerical Results of the Sample Problem 3

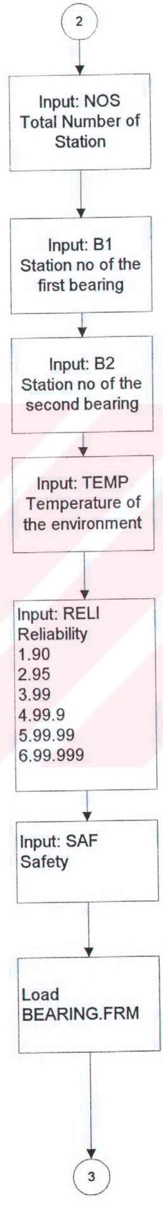
MODE #	CRITICAL SPEED (rpm)	
	Dev. Program	Kelly [29]
1	92138	94758
2	360015	379034
3	780852	852827

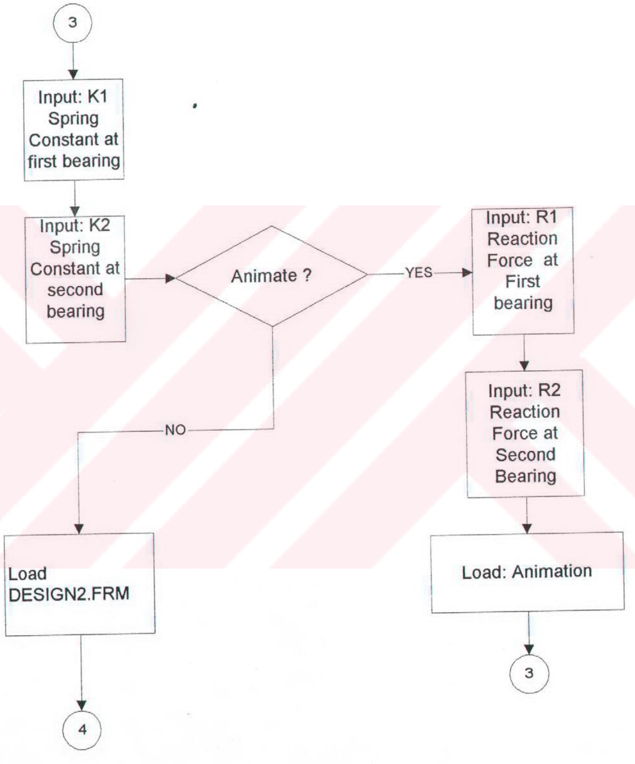
Although formulations that are given in Kelly [29] concerns vibrations of continuous systems, the results that are presented in the above table shows that numerical method that is implemented to the developed program finds the eigenvalues accurate enough for design purpose. It must be remembered that exact results can not be expected for the lumped mass model.

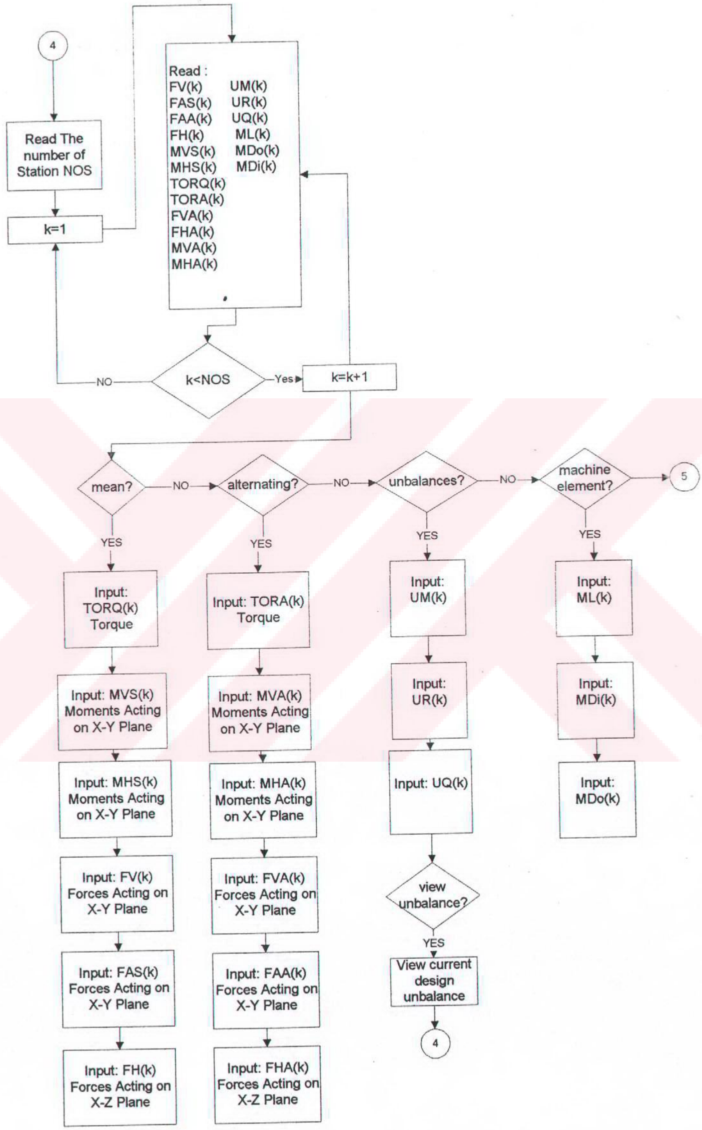
APPENDIX C

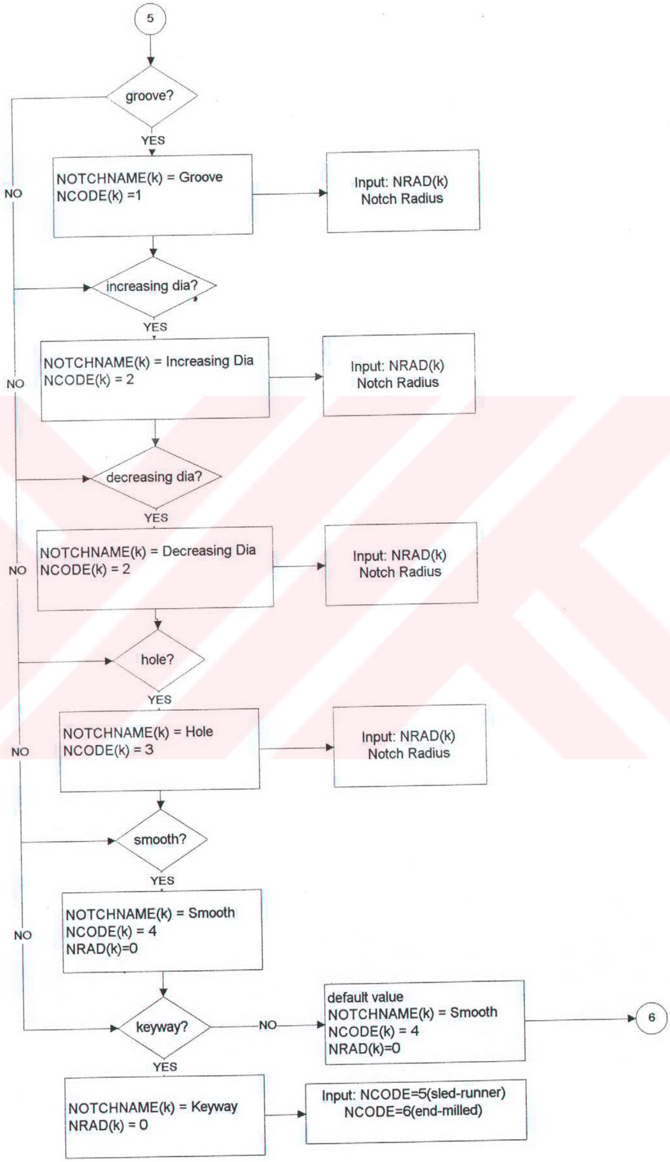
DETAILED FLOWCHART OF THE PROGRAM

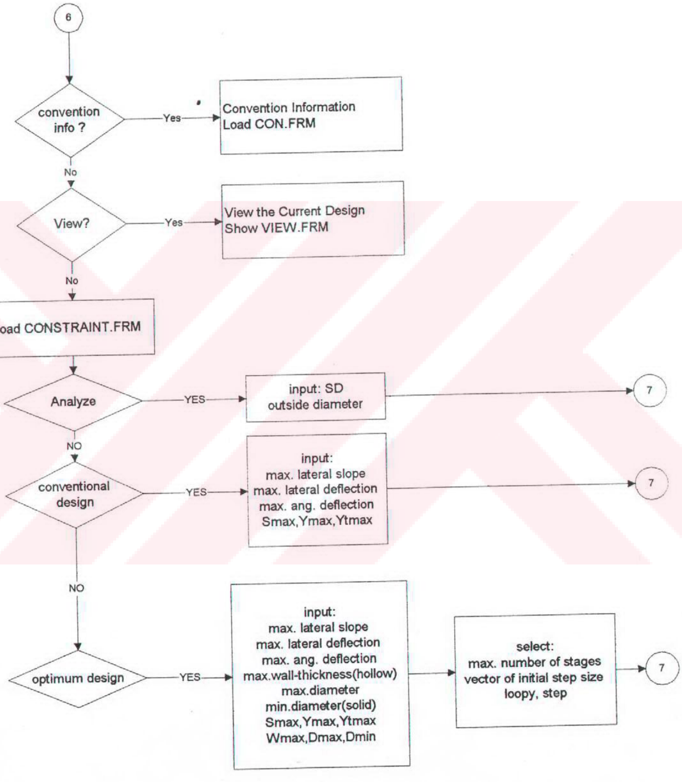


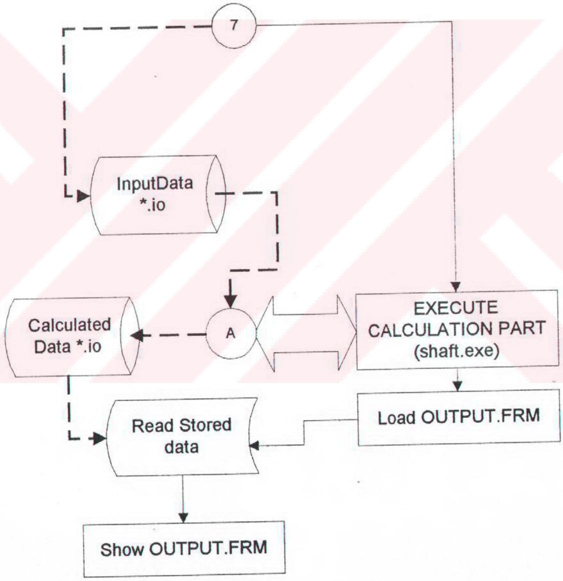

2
Input: NOS
Total Number of
Station
Input: B1
Station no of the
first bearing
Input: B2
Station no of the
second bearing
Input: TEMP
Temperature of
the environment
Input: RELI
Reliability
1.90
2.95
3.99
4.99.9
5.99.99
6.99.999
Input: SAF
Safety
Load
BEARING.FRM
3

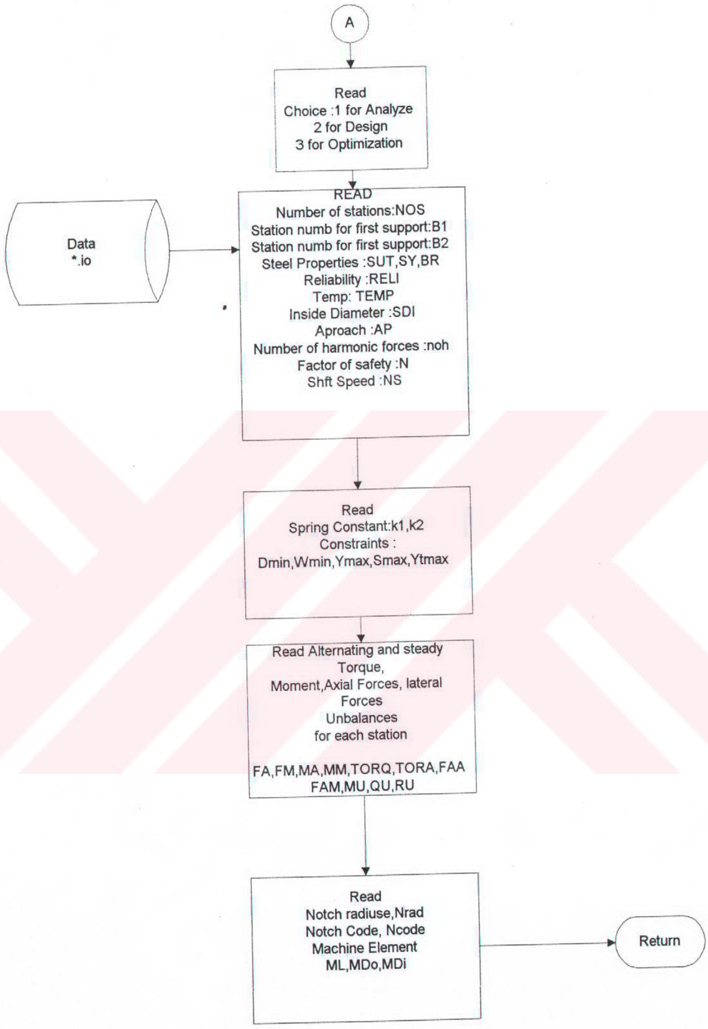

3
Input: K1 Spring Constant at first bearing
Input: K2 Spring Constant at second bearing
Animate ?
YES
NO
Input: R1 Reaction Force at First bearing
Input: R2 Reaction Force at Second Bearing
Load DESIGN2.FRM
Load: Animation
4
3

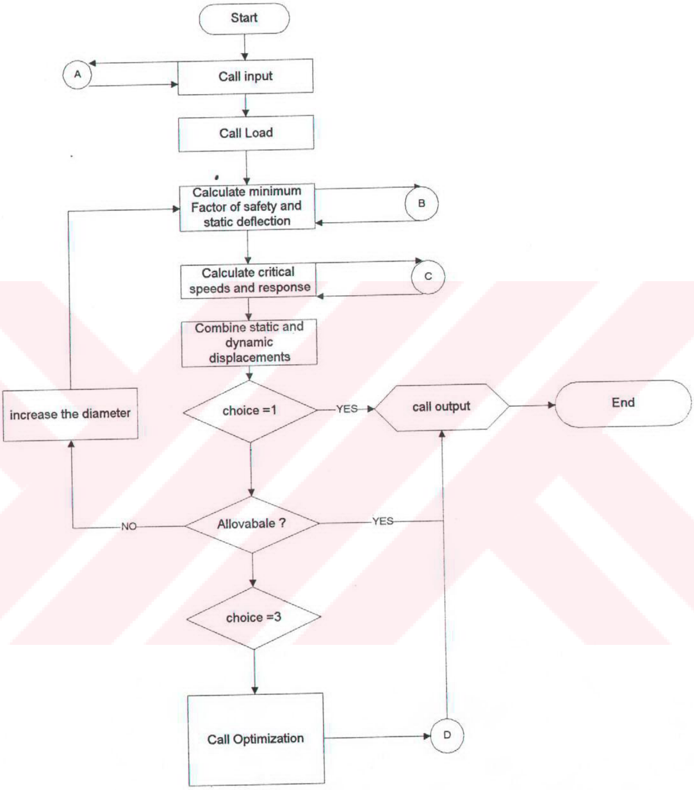

4
Read The number of Station NOS
k=1
Read :
FV(k) UM(k)
FAS(k) UR(k)
FAA(k) UQ(k)
FH(k) ML(k)
MVS(k) MDo(k)
MHS(k) MDi(k)
TORQ(k)
TORA(k)
FVA(k)
FHA(k)
MVA(k)
MHA(k)
k<NOS
NO
Yes
k=k+1
mean?
NO
alternating?
NO
unbalances?
NO
machine element?
5
YES
Input: TORQ(k) Torque
Input: MVS(k) Moments Acting on X-Y Plane
Input: MHS(k) Moments Acting on X-Y Plane
Input: FV(k) Forces Acting on X-Y Plane
Input: FAS(k) Forces Acting on X-Y Plane
Input: FH(k) Forces Acting on X-Z Plane
YES
Input: TORA(k) Torque
Input: MVA(k) Moments Acting on X-Y Plane
Input: MHA(k) Moments Acting on X-Y Plane
Input: FVA(k) Forces Acting on X-Y Plane
Input: FAA(k) Forces Acting on X-Y Plane
Input: FHA(k) Forces Acting on X-Z Plane
YES
Input: UM(k)
Input: UR(k)
Input: UQ(k)
view unbalance?
YES
View current design unbalance
4
YES
Input: ML(k)
Input: MDi(k)
Input: MDo(k)

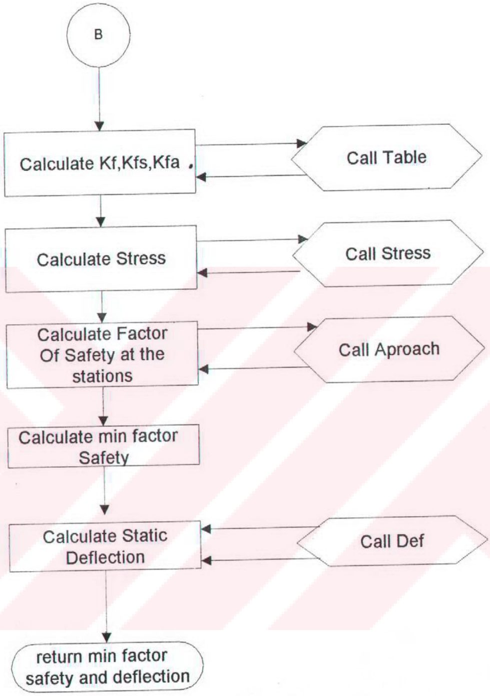

5
groove?
YES
NO
NOTCHNAME(k) = Groove
NCODE(k) =1
Input: NRAD(k)
Notch Radius
increasing dia?
YES
NO
NOTCHNAME(k) = Increasing Dia
NCODE(k) = 2
Input: NRAD(k)
Notch Radius
decreasing dia?
YES
NO
NOTCHNAME(k) = Decreasing Dia
NCODE(k) = 2
Input: NRAD(k)
Notch Radius
hole?
YES
NO
NOTCHNAME(k) = Hole
NCODE(k) = 3
Input: NRAD(k)
Notch Radius
smooth?
YES
NO
NOTCHNAME(k) = Smooth
NCODE(k) = 4
NRAD(k)=0
keyway?
NO
default value
NOTCHNAME(k) = Smooth
NCODE(k) = 4
NRAD(k)=0
6
YES
NOTCHNAME(k) = Keyway
NRAD(k) = 0
Input: NCODE=5(sled-runner)
NCODE=6(end-milled)

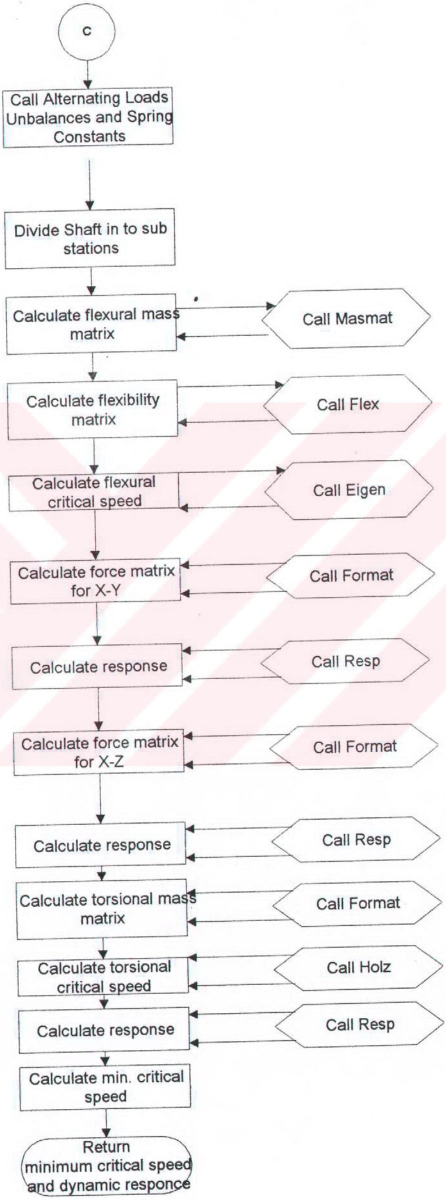

6
convention info ?
Yes
Convention Information
Load CON.FRM
No
View?
Yes
View the Current Design
Show VIEW.FRM
No
Load CONSTRAINT.FRM
Analyze
YES
input: SD
outside diameter
7
NO
conventional design
YES
input:
max. lateral slope
max. lateral deflection
max. ang. deflection
Smax,Ymax,Ytmax
7
NO
optimum design
YES
input:
max. lateral slope
max. lateral deflection
max. ang. deflection
max.wall-thickness(hollow)
max.diameter
min.diameter(solid)
Smax,Ymax,Ytmax
Wmax,Dmax,Dmin
select:
max. number of stages
vector of initial step size
loopy, step
7

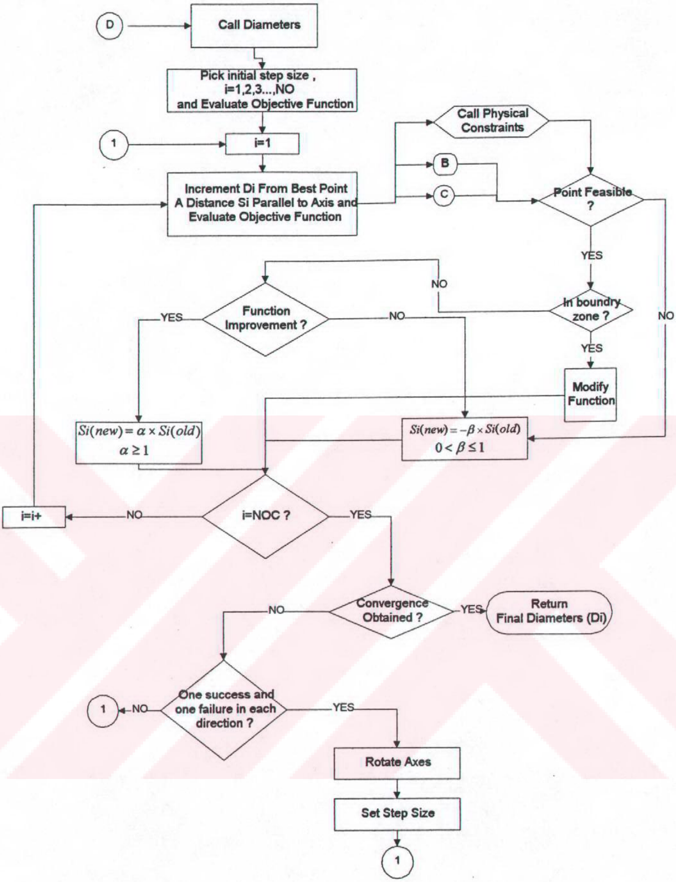

7
InputData
*.io
Calculated
Data *.io
A
EXECUTE
CALCULATION PART
(shaft.exe)
Load OUTPUT.FRM
Read Stored
data
Show OUTPUT.FRM


A
Read
Choice :1 for Analyze
2 for Design
3 for Optimization
Data
*.io
READ
Number of stations:NOS
Station numb for first support:B1
Station numb for first support:B2
Steel Properties :SUT,SY,BR
Reliability :RELI
Temp: TEMP
Inside Diameter :SDI
Aproach :AP
Number of harmonic forces :noh
Factor of safety :N
Shft Speed :NS
Read
Spring Constant:k1,k2
Constraints :
Dmin,Wmin,Ymax,Smax,Ytmax
Read Alternating and steady
Torque,
Moment,Axial Forces, lateral
Forces
Unbalances
for each station
FA,FM,MA,MM,TORQ,TORA,FAA
FAM,MU,QU,RU
Read
Notch radiuse,Nrad
Notch Code, Ncode
Machine Element
ML,MDo,MDi
Return


Start
A
Call input
Call Load
Calculate minimum
Factor of safety and
static deflection
B
Calculate critical
speeds and response
C
Combine static and
dynamic
displacements
choice =1
YES
call output
End
increase the diameter
Allovabale ?
NO
YES
choice =3
Call Optimization
D


B
Calculate Kf,Kfs,Kfa
Call Table
Calculate Stress
Call Stress
Calculate Factor Of Safety at the stations
Call Aproach
Calculate min factor Safety
Calculate Static Deflection
Call Def
return min factor safety and deflection


C
Call Alternating Loads Unbalances and Spring Constants
Divide Shaft in to sub stations
Calculate flexural mass matrix
Call Masmat
Calculate flexibility matrix
Call Flex
Calculate flexural critical speed
Call Eigen
Calculate force matrix for X-Y
Call Format
Calculate response
Call Resp
Calculate force matrix for X-Z
Call Format
Calculate response
Call Resp
Calculate torsional mass matrix
Call Format
Calculate torsional critical speed
Call Holz
Calculate response
Call Resp
Calculate min. critical speed
Return minimum critical speed and dynamic responce


D
Call Diameters
Pick initial step size ,
i=1,2,3...,NO
and Evaluate Objective Function
1
i=1
Increment Di From Best Point
A Distance Si Parallel to Axis and
Evaluate Objective Function
Call Physical
Constraints
B
C
Point Feasible
?
YES
NO
In boundry
zone ?
YES
NO
Modify
Function
Function
Improvement ?
YES
NO
$Si(new) = \alpha \times Si(old)$
$\alpha \geq 1$
$Si(new) = -\beta \times Si(old)$
$0 < \beta \leq 1$
i=i+
NO
i=NOC ?
YES
Convergence
Obtained ?
YES
Return
Final Diameters (Di)
NO
One success and
one failure in each
direction ?
NO
1
YES
Rotate Axes
Set Step Size
1