COMPUTER-AIDED DESIGN OF LATHE SPINDLES

A MASTER'S THESIS

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Middle East Technical University

By

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ABSTRACT

COMPUTER-AIDED DESIGN OF LATHE SPINDLES

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This thesis is on the computer-aided design of spindles for lathes.

A multi-step model of lathe spindles is established. It is constructed by seven parameters : inner diameter, lengths and outer diameters of the span and the overhang, stiffnesses of front and rear bearings. A general formula is developed to determine the radial deflection of spindles. Strength and critical speed calculations are also performed. An interactive program is prepared to automate the design procedure and its use is illustrated by the examples.

Key Words : Computer-aided design, interactive program spindle, radial deflection.

ÖZET

TORNALARIN FENER MİLİNİN BİLGİSAYAR DESTEKLİ TASARIMI

DAİ, Gökhan
Yüksek Lisans Tezi : Makina Müh.Bölümü
Tez Yöneticisi : Yard.Doç.Dr. İ. Hüseyin Filiz
Mart 1985,95 Sayfa

Bu tez, tornalardaki fener milinin bilgisayar destekli tasarımı üzerinedir.

Fener millerinin çok basamaklı bir modeli kurulur. Bu model fener milinin iç çapı, fener milinin basamaklarının dış çapları, rulmanlar arasındaki uzaklık, ön çıkıntı uzunluğu, ön ve arka yatakların rijiditeleri olmak üzere 7 parametreden oluşmaktadır. Fener millerinin radyal sehimini veren genel bir formül geliştirilmiştir. Mukavemet ve kritik hız hesapları yapılmaktadır. Tasarımı otomatikleştirmek için bir karşılıklı etkileşim programı hazırlanmış ve kullanılışı örneklerle gösterilmiştir.

Anahtar Kelimeler : Bilgisayar Destekli Tasarım, Etkileşimli Program, Fener Mili, Radyal Sehim.

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NOMENCLATURE

A_i	area of i^{th} section
a_i	length of i^{th} overhang segment
D_g	pitch circle diameter of the driving gear
D_i	inner diameter of the shaft
D_o	Outer diameter of the shaft
E	modulus of elasticity
g	gravitational constant
I_i	area moment of inertia of i^{th} element
K_m	shear and fatigue factor for bending
K_s	shear and fatigue factor for torsion
K_{sa}	axial stiffness of the spindle
K_{sr}	radial stiffness of the spindle
l_i	length of i^{th} span segment
M	bending moment
M_t	tilting moment at the nose
m	number of overhang segment
N_c	the first critical speed of the shaft
n	number of span segment
P	resultant cutting force in radial direction
P_a	axial component of the cutting force
P_r	radial component of the cutting force
P_t	tangential component of the cutting force
p	number of the span segment which the driving gear on it
Q	resultant driving force in radial direction
Q_r	radial component of the driving force
Q_t	tangential component of the driving force
R_2	reaction force on rear bearing
r	radius of the workpiece, radius of the shaft

r_{io} inner diameter to outer diameter ratio

S_e endurance limit

S_y yield stress of the material

S_u ultimate stress of the material

T torque

W_i weight of i^{th} element

Y_b total radial nose deflection due to the bearings

Y_{bf} radial nose deflection due to the front bearing

Y_{br} radial nose deflection due to the rear bearing

Y_{ov} radial nose deflection due to the overhang

Y_s total radial nose deflection due to the spindle itself

Y_{sa} axial nose deflection

Y_{sm} radial nose deflection due to the P

Y_{sn} radial nose deflection due to the span

Y_{sq} radial nose deflection due to the Q

Y_{sp} radial nose deflection due to the P

Y_T total radial nose deflection

y_i deflection of span at each segment

y_{oi} deflection of overhang at each segment

α factor for buckling

ϕ pressure angle

ρ radius of the curvature

σ bending stress

τ shear stress

τ_d design shear stress

θ_i slope andle at the end of i^{th} segment

δ_{fb} deflection of the front bearing

δ_{rb} deflection of the rear bearing

CHAPTER 1

INTRODUCTION

Spindle units in machine tools perform the functions of holding the workpiece and carrying the forces, and the torques due to cutting and driving forces. Under the action of these forces, spindles exhibit deflections in radial and axial directions which will effect the accuracy of the workpiece produced.

To have an accurate workpiece, especially in NC machines on which the skill of the operator is transferred to the machine itself, the deflections and the deformations of the parts of the machine tool are required to be within specified tolerances. This has necessitated detailed analysis of the parts of the machine tools.

In the case of the spindles, there are a lot of parameters which affect the deflection at the nose. Depending upon the complexity of the spindles in structure, the design invalves many calculations which, in most cases, may be repeated several times to achieve the best solution. This indicates that design procedure is an iterative type.

Availability of fast computers are of considerable asistance in this respect. That is, the repetitive and tedious calculations can be performed by the computers and the designer uses his judgement at different stages of the design.

The aim of this thesis, therefore, is to automate the design procedure and to make the calculations and the plots available to the designer for his final decision.

After surveying on the type of spindles of different lathes a generalised model, in which the spindle is considered as a multi-step shaft, is chosen and the analysis is performed on this model. A generalized formula for the deflection at the nose of the spindle is derived. Strength and critical speed calculations are included for the sake of the completeness of the design.

A computer program (Computer Aided Design Package) is prepared to automate the design procedure. This program works interactively and it is highly user oriented. With the comments inserted into the program, the user is guided at different steps of the design.

The package has the capacity of doing the calculations for a specific spindle and to investigate the effects of the design parameters on the spindle nose deflection. The use of the package is illustrated by an example.

SI units are used throughout the thesis.

CHAPTER 2

LITERATURE SURVEY

2.1. INTRODUCTION

The literature most relevant to the work reported in this thesis is treated in this chapter.

The literature on the design of spindles for lathes and CAD studies on spindles are presented in Section 2.2. Section 2.3 is devoted to the studies on the deflection of antifriction bearings and their computer aided selection.

2.2. DESIGN OF SPINDLES FOR LATHES

KÖENIGSBERGER(1) reported on KIEKEBUSCH's study on the conditions of a main spindle in a lathe headstock. KIEKEBUSCH has observed that, deflection of a spindle depends not only upon its own stiffness, but also upon the inclinations of its bearings under load and, therefore, upon the stiffness of the bearing carryingstructure.

Almost at the same time, BARISH (2) has completed his studies on spindles by measuring deflections in the machine shop. He has investigated the span of spindles and some bearing arrangements by plotting diagrams. He has also obtained the nose deflection by adding deflection of spindle itself and that of bearings separately. He has assumed that, the part of spindle from the front bearing to the application point of cutting force has no deflection, i.e, tilt of the shaft at the front bearing is extended as a straight line. He has considered only a radial cutting force (P) at the cutting zone, and he tried only four span values. He concluded that:

With short distance between bearings (span) the shaft deflections are the least important element. Otherwise, span is the single factor, and particularly the nose bearing should be moved as close to the work as possible. Both bearing deflections are relatively large, and particularly the tail bearing shows more effect than usual.

GJESDAHL (3) has proposed a procedure for determining the deflection of spindles with overhang loads at each end. Straight shafts and shafts which are enlarged between the bearings have been compared for equal slopes of deflection curves at points of support.

HONRATH (4) has measured that the main components in the dispacement of a spindle are the deflection of the spindle(50 to 70 %) and the deformation of the bearing (50 to 30%). He has also observed that with growing load, the rate of deformation is high at first and decreases later. He has explained this result by becoming the radial load more evenly distributed over the rolling elements of the bearings. He uses a radial cutting load in his model. He has calculated the radial deflection of the spindle by summing of deflection of spindle itself and that of bearing. He has suggested that optimum span to overhang ratio lies generally between 3 and 5.

SOKOLOV and FIGATNER (5) have constructed a model for radial deflection of spindles. Their spindle model is a one-step, solid shaft with two supports and has stiffnesses of the span and the overhang separately. They have calculated the rigidity of the spindle at the cutting point by considering the rigidity of the shaft itself, effect of the deflection of front bearing and that of rear bearing due to a radial cutting force at the nose of the spindle. They have proposed a method of calculation by considering the type of spindle bearing, distance between bearings and length of overhang as the basis of design factors. They have constructed nomograms for two different series of bearings to find the optimum values of spindle diameter and span to overhang ratio for a given value of nose stiffness. They have stated that the stiffness of the spindle should not be less than 25 000 kg/mm (245 000 N/mm) for standard accuracy machine tools, and at least 50 000 kg/mm (490 000 N/mm) for high precision machine tools. They have also prepared a diagram for a few types of bearings in Soviet standards to show their relation of radial stiffness vs inside diameter.

BOLLINGER and GEIGER (6) have developed a method of analysis for prediction of the static and dynamic behaviour of machine tool spindle systems. A mathematical model has been prepared by use of the finite difference technique, and a general purpose analog computer has been utilized to solve the equations in the model. They have constructed the spindle as a shaft with 10 discrete masses. The influence of bearing stiffness, bearing position, effect of third bearing, damping, a time dependent radial force at the tip, and the presence of the workpiece have been discussed. They have modelled the workpiece as a cantilevered, 3 equal masses at the spindle nose. They have obtained the stiffness of bearings by experiments on the existing lathe and have concluded that the use of three bearings gives no appreciable improvement over the optimum position of two bearings. They have verified their analog computer model by measuring the deflections on the lathe. They have found 8 % difference between calculated and measured first mode resonant frequency of the spindle.

SHUZI (7) has discussed a number of factors on radial static stiffness by using the concept of "influence factors". He has pointed out the importance of driving force (Q) and its location, and couple moment (M_t)produced by axial component of cutting force besides the importance of cutting force (P). He has studied also on the number of bearings (radial and thrust bearings) and their mounting positions. He has investigated the optimum span of a spindle mounted in two or multiple bearings by using nomograms. He has developed direct and indirect influence factor methods for calculating the reaction of any bearing and the displacement of any position for an elastic beam possessing many elastic bearings. He has discussed also the problems of uncoaxiality of the housing holes for a spindle possessing many elastic bearings. He has concluded the following:

1- Relavent structure and parameters must be properly chosen by considering the effects of P, M_t and Q at the same time for different operating conditions.

2- Shortening the overhang, increasing the area moment of inertia of the spindle and the stiffness of the front bearing are effective measures of enhancing the static stiffness of the spindle.

3- Whether or not the static stiffness of a spindle can be improved by increasing the number of supporting points depends on whether it is P, or M_t, or Q that is exerted on the spindle. For the case of P being exerted, it will be increased; for the case of M_t and that of Q exerted on the overhang, it will in general be increased, while for the case of Q acting on the span or on the tail, it can be increased or decreased, or it may remain unchanged.

4- For the case of P, for an optimum span, the spindle will have considerable stiffness and the effect of the third bearing is insignificant. If span is much greater than the optimum span and a middle bearing is used at a distance to the front bearing approximately equal to the optimum span, then the static stiffness will be greatly increased.

5- It is much more desirable for the thrust bearing to be mounted in the front than in the rear bearing. If stiffness of front bearing and span diameter to span ratio are both large enough, then the influence factor of the spindle nose approaches that of the overhang beam with a fixed end no matter whether the spindle is under the action of P, or M_t or Q.

Axial stiffness of spindles has been studied by BORSHCHEVSKII and et al (8). They have stated that the axial deflection of a spindle may be computed by general deformation formula and contact stiffness formula. They have investigated the axial stiffness of various thrust ball-bearings in Soviet standards.

PEROTTI (9) has investigated the flexural vibration of lathe spindles. He has divided the spindle under investigation into five parts, considering the driving gear on the spindle as a part of it. He has used an analytical-graphical approach, as is used for finding the critical speed of a shaft. Then, he has compared the measured, actual (free and damped) oscillation value with the calculated free and undamped one, and he has concluded that there is no considerable difference between them.

EL-SAYED (10) has prepared an n-stepped spindle model with two points of support for digital computers. He considers only the radial cutting force at the tip of the spindle. He has prepared some diagrams to see the effects of some parameters of the spindle. He has fixed some parameters and has watched the deviation on total stiffness of spindle or an optimum span due to the other parameters like front and rear bearing stiffnesses, overhang, inertia of span. He has concluded that, the span inertia factor had a negligible effect on spindle nose stiffness, the nose stiffness of a spindle of constant diameter was a good approximation for its equivalent multi-stepped spindle; and increasing the stiffness of the front bearing produced a marked increase in spindle nose stiffness, but increasing the stiffness of the rear bearing had little effect.

Another computer-aided design model for spindles has been realised by STANSFIELD (11). He has investigated the spindle analysis in two computer programs. First program analyses the spindle by seven parameters (inner diameter, front outer diameter, rear outer diameter, front and rear bearings stiffness, overhang and span) to make accelerated the optimisation process. The second one is a multi-stepped model and it gives precise deflections and deflection-gradients at the bearings and at the cutting zone. No result has been declared in the study.

2.3. ANTIFRICTION BEARINGS

One of the several books an rolling bearing analysis is belong to PALMGREN (12). He has developed a set of formulae for calculating the deflections of all common types of bearings subjected to radial or axial loading. They are based on Hertz's theory. He has assumed that contact angles areconstant and the bearings are geometrically perfect.

HARRIS (13) has also performed a valuable study on deflection of bearings. Furthermore he has considered the effects of shaft, housing fits and the change in contact angles on deflection calculations.

JONES (14) has produced a general theory for elastically constrained ball and roller bearings under arbitrary load and speed conditions. Shaft and housing elasticities has been considered as well as centrifugal and gyroscopic loading of the rolling elements under high speed operation.

FIGATNER (15) has developed a method of calculating the deflections of thrust and angular contact ball bearings. He has used Hertz's theory in these calculations and he has investigated the geometrical errors in bearings and the effect of preloading. He has concluded that, the reduction in manufacturing errors yields higher rigidity.

SCHREIBER (16) and KUNERT (17) have calculated the deflections of single and preloaded bearings. They have considered the effects of radial clearance and change in contact angle but they have neglected the geometrical errors and elasticity of the mounting.

FILIZ (18) has derived a formula for axial stiffness of bearings which is quite flexible for digital computation. It calculates the stiffness of bearing arrangements in terms of load-deformation constant, preload, and number of bearings on each site of the arrangements. His formula neglects the geometrical errors and assumes constant contact angle under load but it gives only 0.85 % error.

SWANSON (19) has treated two preloaded bearings as two parallel springs, so that the resulting stiffness has become twice as that of one. On the other hand, he has suggested that high preloads had dangerous effect on bearing life.

LEIBENSPERGER (20) has investigated the effects of preload and end play on the dynamic behavior of a rotating machine. He has developed a technique to determine the specific values of preload for a particular design to optimise the dynamic characteristics of the machine.

A recent study has been published by LEVINA (21). He has suggested formulae for calculating the radial and axial stiffness of angular-contact ball; angular-contact thrust and taper roller bearings. He has constructed additional graphs to determine the coefficients in these equations. He has also derived a formula for restraining moment, created in angular-contact thrust ball bearings.

Another person who has studied the influence of preloading is HONRATH (4). He has reached the result of decreasing in radial deflection of bearings with preloading according to his measurements. He has stated that, preloading reduces the spindle nose deflection further due to the clamping moment which the bearing exerts on the spindle.

OPITZ, GÜNTHER, KALKERT and KUNKEL (22) have also studied on deflection of bearings. They have investigated the effects of radial play, manufacturing errors, bearing housing, and operating conditions. They have concluded that a reduction in positive play, caused by the temperature gradient between the inner and outer races, increases the initial stiffness. They have also given a nomogram to find deflection of bearings of type NN30 and NNU49.

ELLIS (23) has constructed a computer-aided selection program for rolling element bearings. The criteria used in the selection are only the bearing life and static strength. He has not considered the mechanical requirements in his programs.

FILIZ (24) has described also an interactive computer aided selection program for antifriction bearings. His main concern is on the selection of bearings for leadscrew assemblies, but the program automates also the selection of bearings for general machinery applications. The program states that the selected bearing is sufficient or not for the conditions given, due to the load carrying capacity and life requirements. Program is capable of bearing selection for single site or for double sites. It includes preloading requirement and further, a formula has been derived in the study for preloading. Program consists also of axial deflection (stiffness), bearing inertia, and frictional torque calculations and lubrication selection.

Another computer-aided selection procedure for antifrication bearing has been developed by ARIKAN (25). For selection process, rotational speed, minimum required life, reliability, operating temperature, minimum static factor of safety, arrangement of and preference on bearings are the requirements. Preferred bearing types with preferred oils or grease and their performance characteristics such as, minimum and maximum bearing life, friction torque and power loss, lubricant viscosity at operating temperature are obtained and tabulated.

CHAPTER 3

INITIAL CONSIDERATIONS ON CAD OF SPINDLES

3.1. INTRODUCTION

This chapter is included to give an overall view of the spindles and of the CAD structure.

Practical examples of the spindles are presented in Section 3.2. Section 3.3. is devoted to the general requirements of the spindles. The role of the computers in the design process and the view taken in CAD of spindles are discussed briefly in Sections 3.4 and 3.5. CAD structure is discussed in Section 3.6.

3.2. SPINDLES

In turning, the main spindle serves for centering and holding the workpiece under the effects of weights and cutting forces on the one hand and driving forces and torques on the other. It fulfils, therefore, two functions, i.e. it does not only locate the workpiece but also drives it with the required accuracy.

The accuracy of the workpiece depends on the size and the layout (configuration) of spindles. As illustrated in the following examples, each configuration has its own merits.

In most of them, bearings are arranged in such a way that axial force is carried by axial thrust bearings and radial load is carried by radial bearings. Figure 3.1

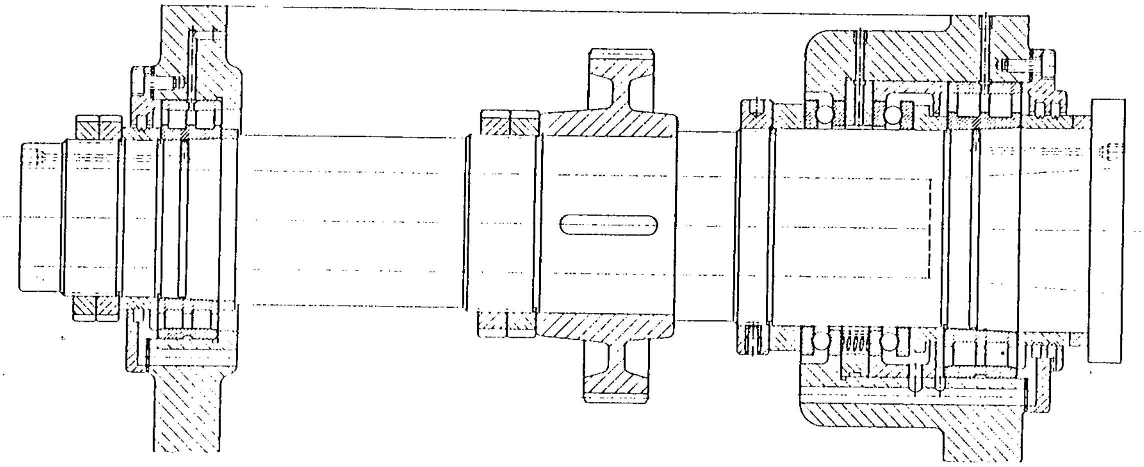


Fig.3.1. A Lathe Spindle (Ref.27)

shows such an application. The radial bearings are double row cylindrical roller bearings. The two roller rows ensure a high bearing load capacity and rigidity. Axial location is provided by two single-direction thrust ball bearings. All bearings are separable which simplifies mounting and dismounting.

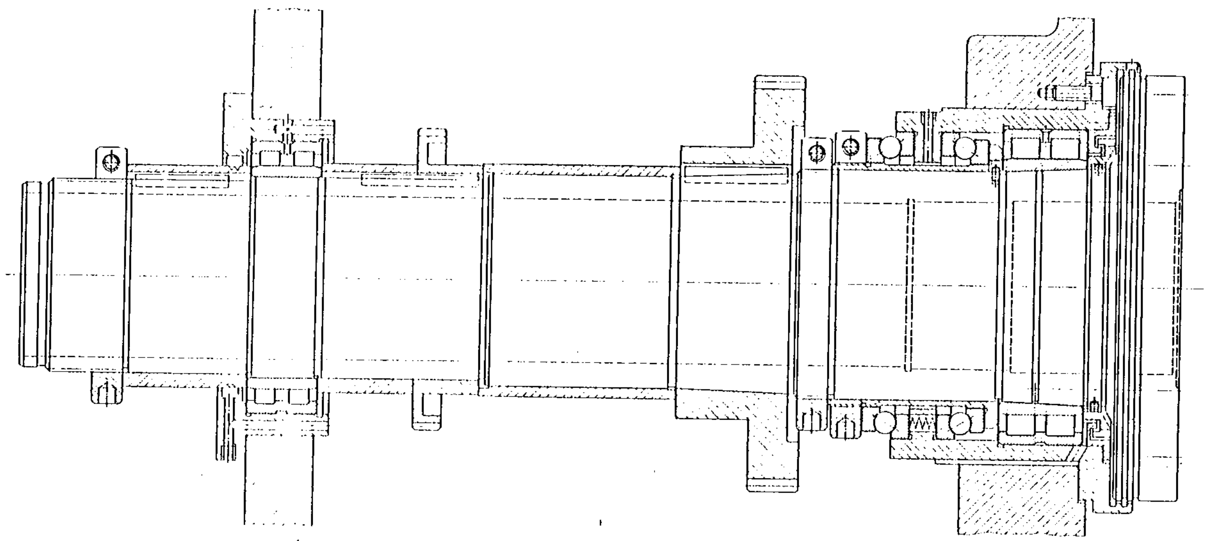


Fig.3.2. An NC-Lathe Spindle (Ref.27)

In Figure 3.2 a similar arrangement has been employed on an NC-Lathe spindle. The Spindle diameter is however, larger so the circumferential bearing speeds are considerably higher. Thus, no thrust ball bearings, but two angular contact thrust ball bearings have been used. These bearings have a contact angle of 60°, thus, they are able to accommodate the radial ball centrifugal forces more favourably than thrust ball bearings.

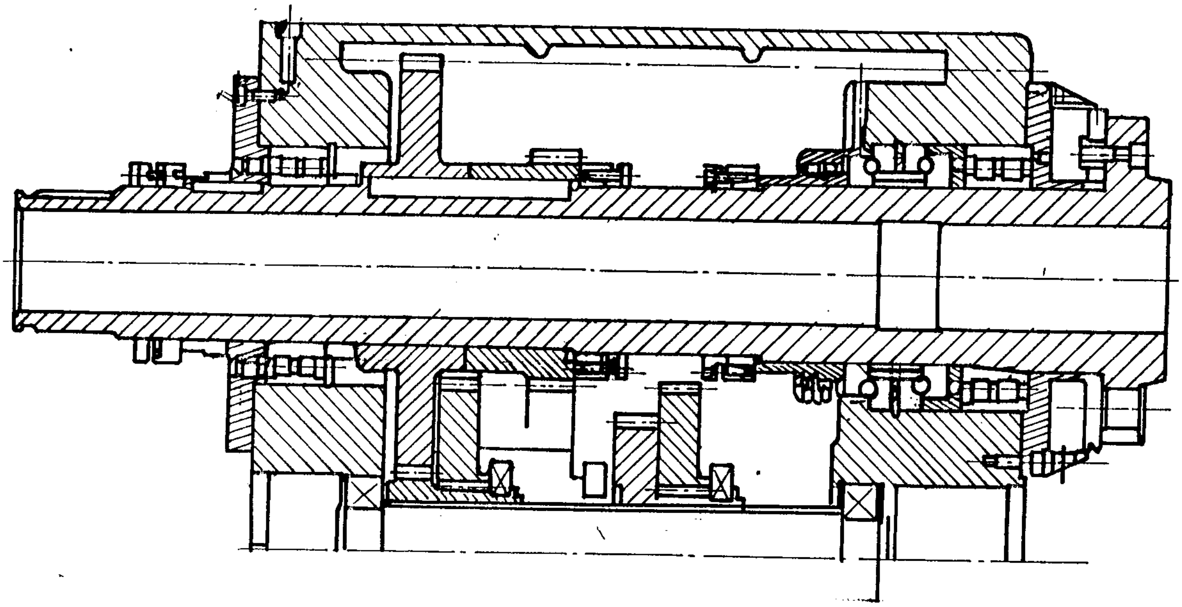


Fig.3.3. Spindle for CNC-Universal Turning Machine(DN/DNE)

Traub has used similar arrangement as shown in Fig. 3.3 in CNC-Universal turning machine DN/DNE. Angular-contact thrust ball bearings are used to accommodate the axial load.

Tezsan has established another design with a radial roller bearing at the work end and two separately preloaded angular-contact ball bearings at the opposite end. This is shown in Fig. 3.4.

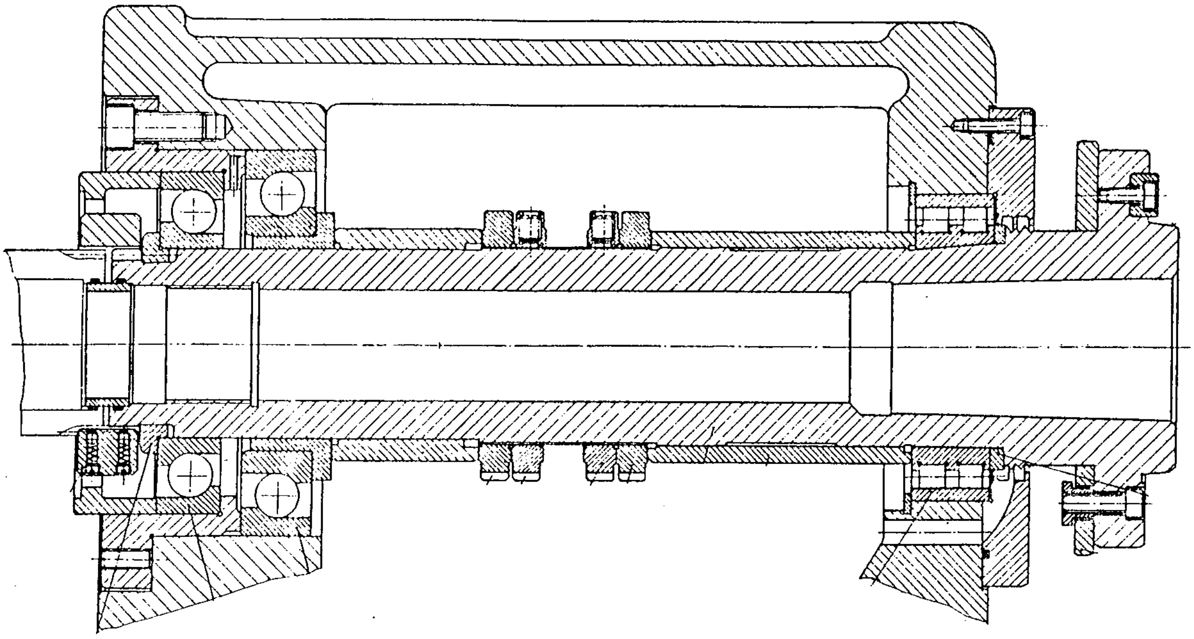


Fig.3.4. Spindle of The Lathe Mode by TEZSAN (Ref.29)

It may be possible to reach higher velocities by using angular-contact ball bearings throught the design. SKF (30) and FAG (31) produce high precision angular-contact ball bearings especially for spindles. They have high axial stiffness and high axial load carrying capacity due to contact angles of 15^{o} or 25^{o}. They may be used in arrangements of back-to-back, tandem, face-to-face or any combinations of these. Bearings arranged back-to-back provide a relatively stiff bearing arrangement which can also accommodate tilting moments.

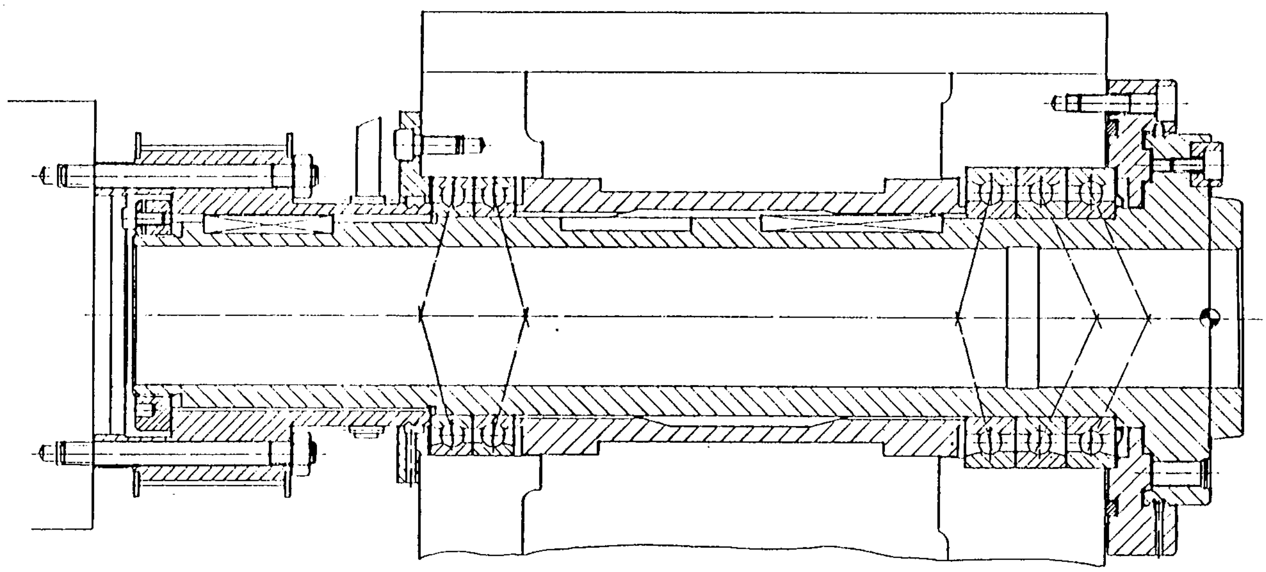


Fig.3.5. Spindle For CNC-Lathe (TNS) (Ref.32)

Fig. 3.5 shows such an arrangement performed by Traub on CNC-Drehautomat, of TNS types. The angular-contact ball bearings are mounted in a combination of tandem and back-to-back arrangement at the work end and in back-to-back arrangement at the rear end.

The design can often be simplified considerably by the application of taper rollar bearings.

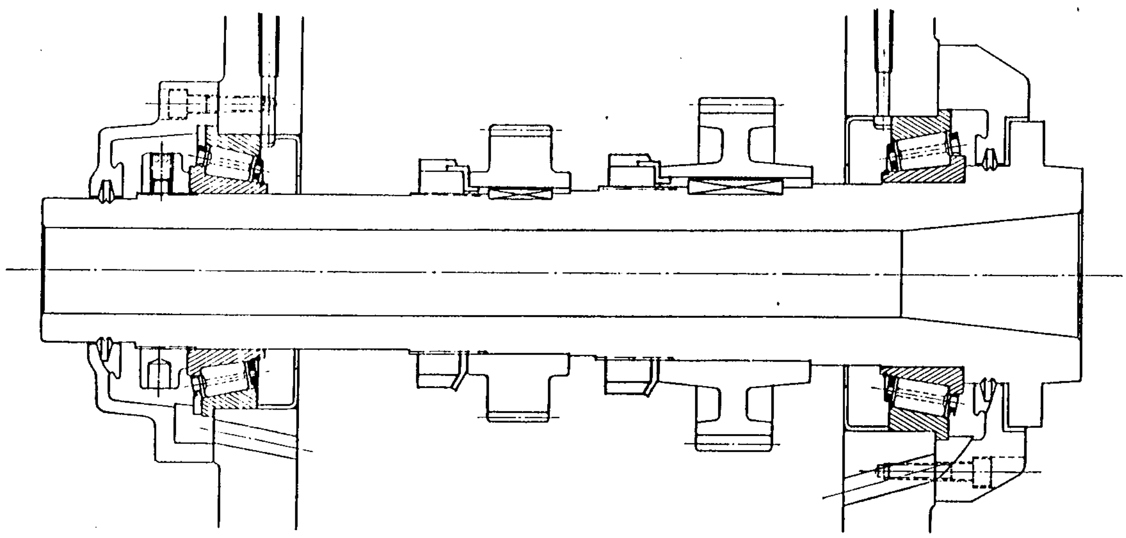


Fig.3.6. A Lathe Spindle (Ref.33)

Spindle design, in Fig. 3.6 is very simple and of low cost, but the interference fits, which must be used on the cups by reason of the unfavourable width to diameter ratio can easily distort the cup tracks if housing bore roundness is not perfect.

Fig. 3.7 shows a spindle mounted on a double-row tapered roller bearing at the work end and a single row tapered roller bearing at the rear end. It will be seen that a pre-loading, on the bearing at rear end, is achieved by means of springs. This method has been developed by Precision Industrielle for ensuring a constant pre-set value of preload whilst accommodating spindle expansion which may occur due to temperature rise.

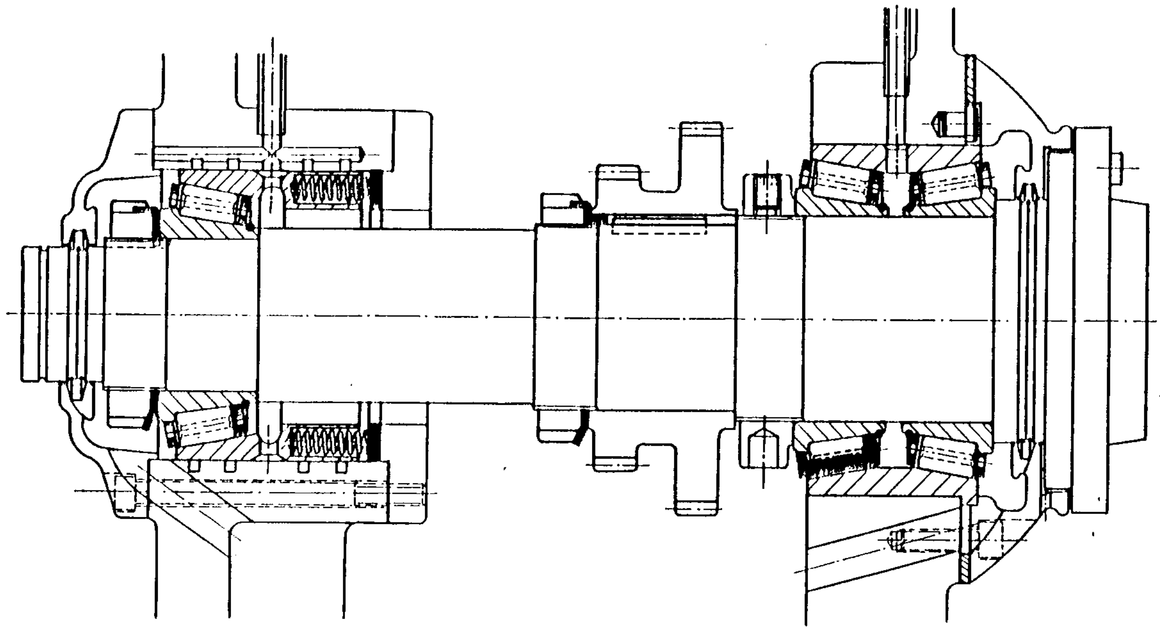


Fig.3.7. A Lathe Spindle (Ref.33)

Another arrangement of tapered roller bearings is shown in Fig . 3.8. Two preloaded tapered roller bearings, arranged in back-to-back, take axial load and a part of the radial load while cylindrical roller bearing at the rear end takes only a part of the radial load.

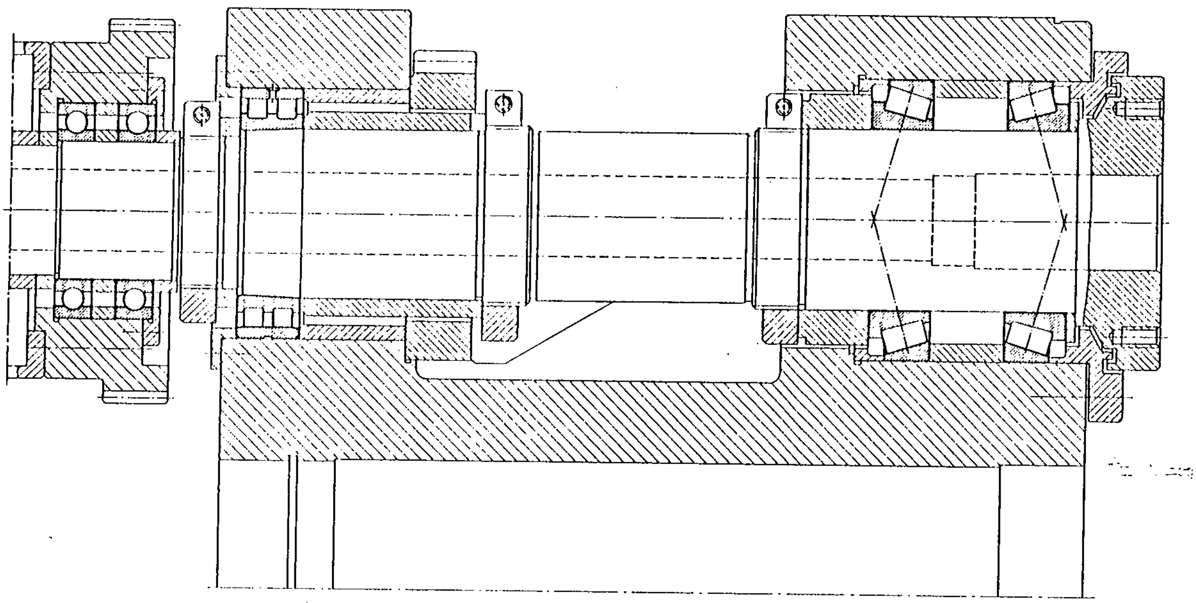


Fig.3.8. A Lathe Spindle

- The addition of a third bearing between the two existing bearings may also provide a reduction in the static deflection of the chuck. But it brings together the problems of manufacturing and cost. In fact, the studies (1), (26) have shown that the effect of the third bearing on the static stiffness of the spindle is insignificant, if the span between the front and the rear bearings is optimum span. Fig. 3.9 shows a spindle with three bearings.

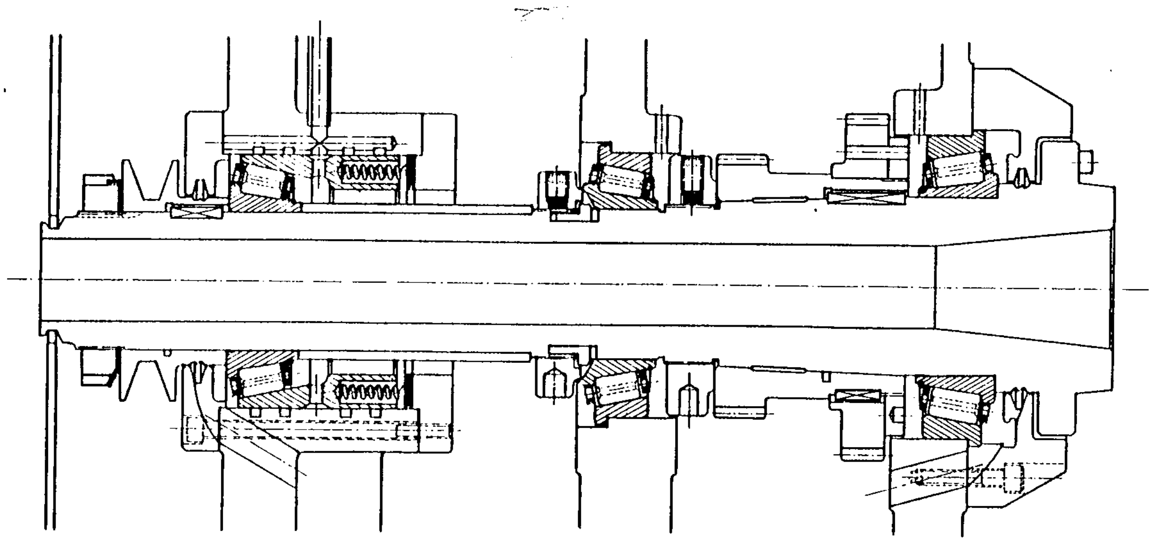


Fig.3.9. A Lathe Spindle (Ref.33)

It is seen that, almost all the spindles in the examples above are multi-stepped shafts. This result comes from the fact that, the bearings and gears on the spindle require shoulders to restrain for mounting and preloading purposes. It is also worthwhile to observe that the diameters of those steps become greater gradually from tail to nose. Further , the spindles have constant diameter bores, except for a slightly conical shape at the nose.

3.3. REQUIREMENTS OF SPINDLES

The aim of a machine-tool designer is to obtain the specified accuracy of shape of the workpieces produced on the machine. The machining quality depends upon various factors such as: The tool (shape and material, rake angles, quality of cutting faces), the tool carrier (stiffness of milling arbor or boring bar, etc., quality of tool clamping), the workpiece and its clamping (machinability of the material, stiffness of the workpiece and of the clamping fixture, accuracy of centers, etc.,) the selected cutting conditions (cutting speed, depth of cut, feed rate), and changes of working conditions which may occur during the operations and may be caused by the machining processes (tool wear and crater formation, temperature changes, etc.). But the most leading factor in machining quality is the machine-tool itself. Hence, careful considerations have to be given in the design of the parts of the machine tools.

Spindle is one of the most important units which influences machining accuracy of the machine tool. This fact imposes the following principal requirements on the spindle unit of machine-tools:

Rotational accuracy : It is usually characterized by the runout of the nose (front end) of the spindle.

Rigidity : Rigidity is determined as the capacity of the spindle to retain its correct position when acted on by various working forces. Excessive deformation of the spindle has a detrimental effect on the machining accuracy and on the service life of the spindle bearings.

Vibration-proof properties : They should be possessed by the spindles of high speed machine-tools, especially for performing finishing operation.

Wear resistance : Wear resistance of the bearing surfaces is required in cases when the spindle runs in sleeve bearings.

These requerements are satisfied by correctly selecting the design parameters of the spindle. Those parameters are:

Inner and outer diameters of the spindle itself, the distance between the bearings (length of span), length of overhang part, the stiffnesses of the bearings, the position of the point of application of the driving force.

3.4. THE ROLE OF COMPUTERS IN DESIGN

Often design problems can be treated by what Offner has called "Brute Force" optimization. In this approach, an important input parameter is varied in a discrete manner, each new value producing a new solution and each solution representing a modified system. Thus a search is made for an admissible solution which satisfies some important performance criterion or criteria.

Computers are effective devices in such an iterative approach. They can perform the routine, tedious and repetitive calculations in a much shorter time. They are capable of making very complex calculations in a great assurance. Nevertheless, there are some requirements which must be satisfied in order that the potential economic usefulness of computers in design can be fully realized. Computer programs must be highly user-oriented with respect to data preparation of results and the ease with which they might be integrated into the normal processes of design.

3.5. THE VIEW TAKEN OF THE CAD PROCESS

The design procedure of spindles consists of two steps: The first, is the sizing of the spindle and the second is the selection of the most suitable bearings. As illustrated in Section 3.2, there exist a large variety of spindle designs with different types of bearings at different arrangements.

The spindle is rarely a single-step shaft. In most applications, it is a multi-step shaft and sizing and deflection calculations involve a great deal of computational work. Furthermore, optimization of the parameters of a spindle requires tedious calculations repeated many times.

In view of the above discussion, the design procedure is iterative. As mentioned in the previous section, fast computers with interactive facilities are of considerable assistance in this respect. With highly user-oriented computer programs, it is possible to obtain an optimum design of the spindles.

3.6. CAD STRUCTURE FOR SPINDLES

The proposed model for the spindle of lathes in this study is a multi-stepped, hollow shaft with two points of support. It is shown in Figure 3.10.

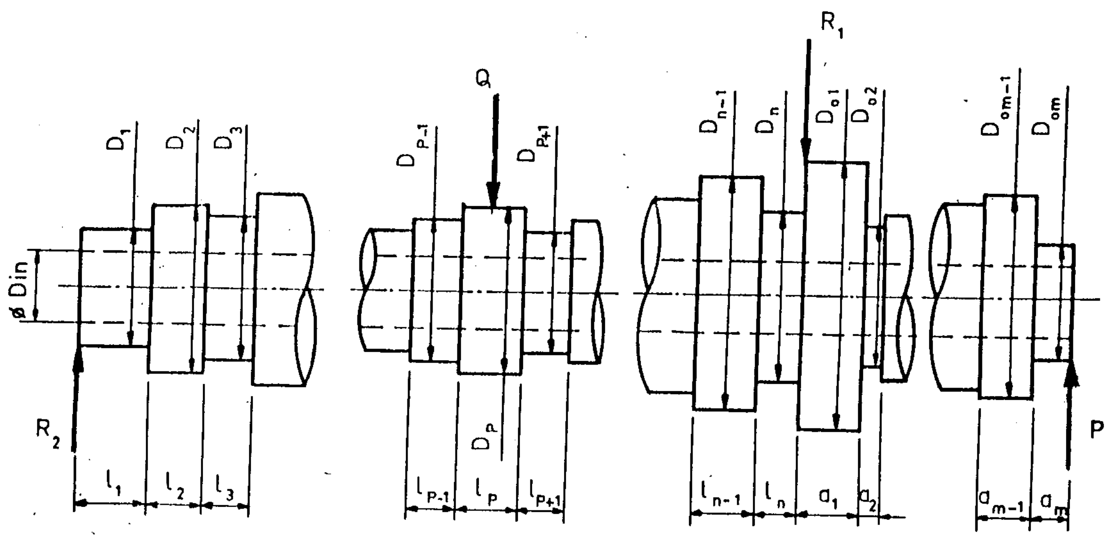


Fig.3.10 The Proposed Model for the Spindle of Lathes.

The spindle (shaft) has a driving gear on its span so, it consists a radial driving force on the span in addition to the effect of cutting force at the nose. It will also be possible to consider the tilting moment at the tip due to the axial component of the cutting force by this model. Chuck and workpiece may be considered as an integral part of the spindle in this model.

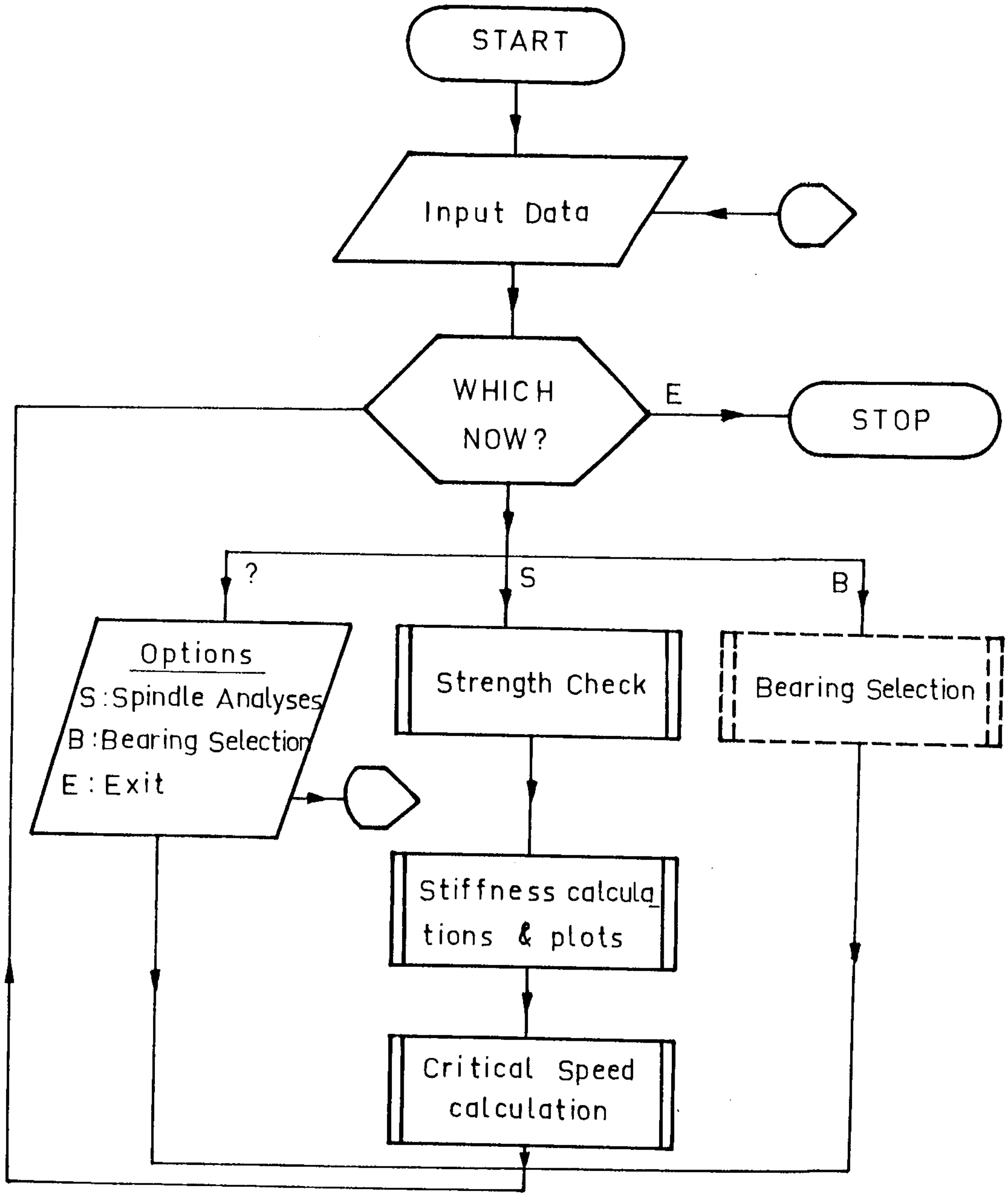


Fig.3.11. Simplified Flowchart of the Computer Program.

A computer program has been prepared to design the spindles of lathes by using this model. Its simplified flowchart is shown in Figure 11.

The analysis part calculates the axial and radial deflections of the spindle, plots graphs to see the effects of selected parameters to reach a better design at each step and checks the spindle for strength and for critical speed. The bearing selection part is not included in this study but, if it is desired to see the effects of bearing deflections on the performance of spindles, the stiffnesses of the bearings are to be inserted into the analyses part as the input. It is proposed to use this program in connection with one of the computer-aided bearing selection programs stated in the previous chapter after adaptation.

The program has interactive characteristics. With the questions and the comments inserted into the program the user (designer) is guided to reach a better design among several alternatives.

The program can be run by any computer with the interactive facilities. The results in this thesis are taken by using facilities of Computer Center at Gaziantep Campus of the Middle East Technical University. The computer hardware structure of this center is shown in Figure 3.12. The program is written in Fortran Language.

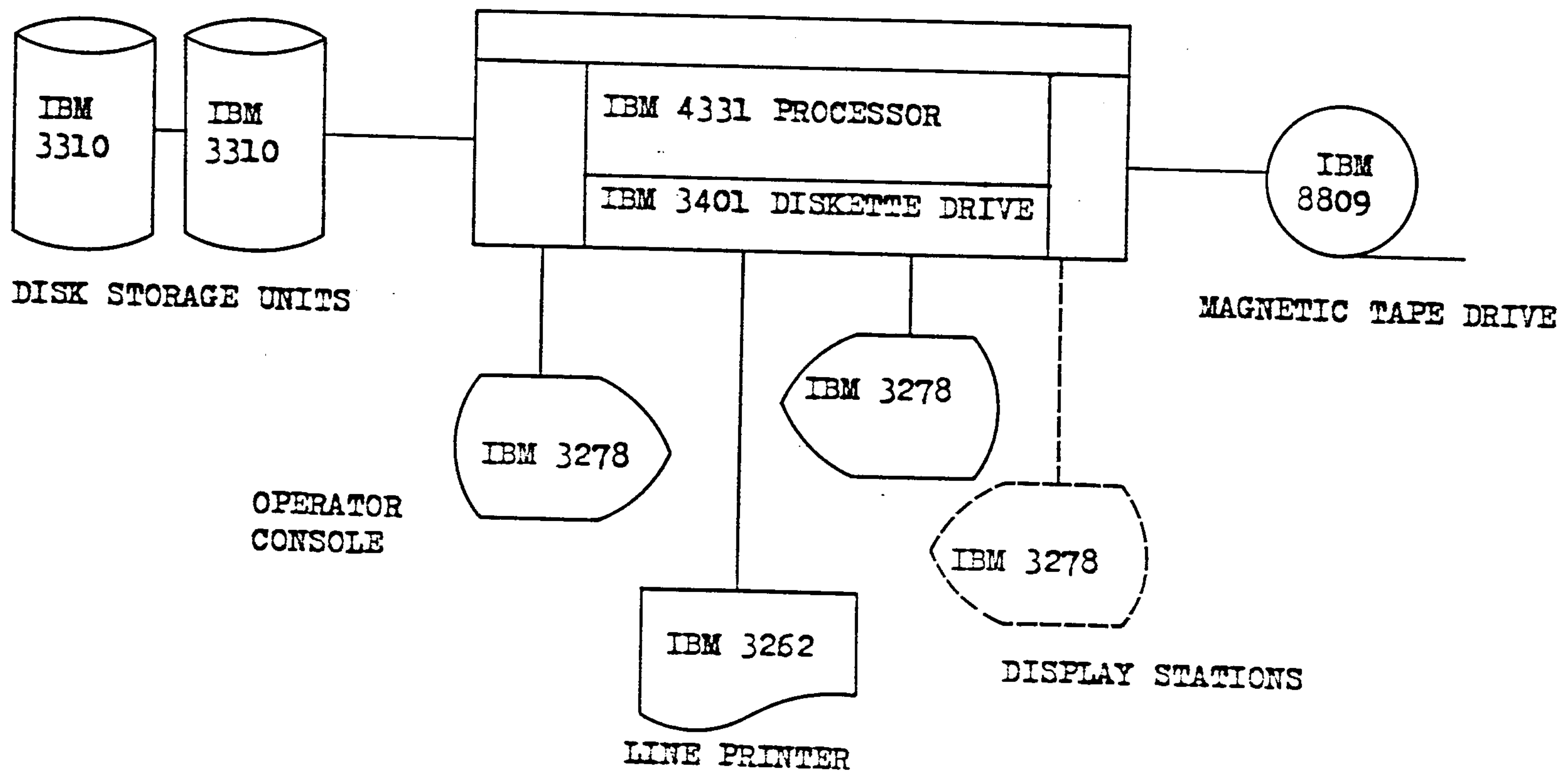


Fig.3.12. The Computer Facilities at Gaziantep Campus of the Middle East Tech. Univ.

CHAPTER 4

DESIGN OF SPINDLES

4.1. INTRODUCTION

In this chapter, design of spindles on the basis of stiffness and strength criteria is presented.

Although materials of the spindles are beyond the scope of this study, it is considered worthwhile to mention about the materials in Section 4.2, which is extracted from Reference (33). Cutting and driving forces analysis is introduced in Section 4.3. Section 4.4 gives nose deflection for multi-stepped spindles. Design of spindles by strength and calculation of critical speed is given in Section 4.5 and 4.6 respectively.

4.2. SPINDLE MATERIALS

A variety of alloy steels are currently in use for the manufacture of machine-tool spindle shafts.

Nitrided steel will give excellent results since the low hardening temperature produces a very stable component. However, the initial material cost, coupled with the cost of accurate turning and grinding before heat treatment, may be lost if eccentricity on finish grinding results in the penetration of the thin hard skin.

When using nickel-chrome and other alloy steels every care should be taken in its heat treatment to avoid the necessity of straightening operations, since it becomes practically impossible to stress-relieve the shaft sufficiently to prevent warping after each grinding operation. The normalising process as recommended by the steel supplier should be carried out and the use of salt bath preheating and quenching will further reduce the possibility of distortion.

A stabilising operation should be introduced immediately before finish grinding since it is important in alloy steels that the austenitic transformation is entirely completed. Stabilisation by deep freeze or tempering method may be used, the latter is preferred.

A medium carbon steel which is a comparatively "dead" material may often be used with advantage, since distorsion can be kept to a very low level and superficial hardening can be arranged by induction or flame heating at the appropriate points. They do not normally require stabilising operations.

4.3. CUTTING AND DRIVING FORCES IN TURNING

4.3.1. Cutting Forces

The metal cutting process is a result of two relative movements between the cutting tool and the material which has to be machined. The "cutting movement", i.e. the relative movement between cutting edge and workpiece material, results in an amount of metal corresponding to the depth of cut being separated from the workpiece material in the form of chips, the "feed movement" brings new material in front of the cutting edge after a particular cut has been completed. The resistance caused by deforming the workpiece material and by frictional forces acts in the form of a cutting force on the tool (action), and on the workpiece (reaction). The various parts of the machine tool (the structure, slideways, spindles, workpiece and tool carriers, etc.), must be able to carry the resulting loads, and the driving elements must transmit the corresponding forces and torques at the required velocities.

A knowledge of the forces and velocities which occur during the various cutting processes is the essential basis for determining the size and material of the load transmitting elements together with the required driving power, that is, for the design of machine tools. However, the machine tool designer need only know the order of influence of the various parameters and those results of research work which are essential for the actual design of the machines.

The cutting force is not constant, but pulsating even during simple turning operations. This is due to the elasticity of the tool, the workpiece and the machine, to the resulting changes in depth of cut, the effective rake angles, the relative velocity between tool and workpiece and to the chip formation itself. A hard spot in the workpiece material, for instance, can produce elastic deformations in tool, workpiece and maçhine and thus initiatesa vibration. For a machine tool designer there is little value in discussing the size of cutting load, because practically every machine tool must accommodate widely varying conditions. Hence, for each case, an arbitrary load is selected and a location approximating the range of conditions under which the machine must do its best work. For that reason any calculation can only be comparative in means of determining relative values of various design factors under an arbitrarily selected load condition .

The cutting force which acts upon the tool is conveniently resolved into components in three directions as in Figure 4.1.

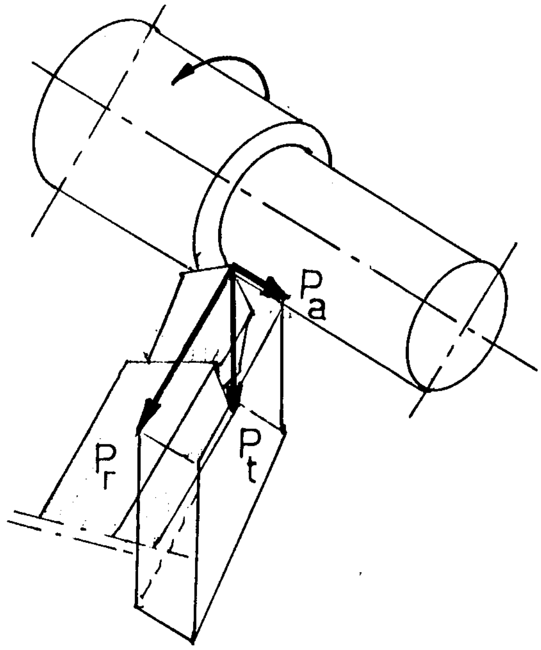


Fig.4.1. The Cutting Force Component in Turning.

1- Tangential component is tangent to the turned surface, and at right angles to the axis of turning, i.e., it is in the direction of the cutting speed. This is the main component (P_t) which together with the cutting speed determines the net power required for the main spindle drive.

2- Axial component is parallel to the axis of turning, i.e., it is in the direction of the feed movement. This is also called as the feed force (P_a), which together with the feed velocity determines the net power required for the feed drive.

3- Radial component (P_r) is radial to the turned surface, i.e., it is in the direction of the depth of setting movement.

The tangential component (P_t) and the radial component (P_r) may be added vectorially to obtain a single resultant radial cutting force component (P)

$$P = \left[(P_t)^2 + (P_r)^2\right]^{1/2} \qquad (4.1)$$

According to Koenigsberger (1), the radial component (P_r) can be assumed as $0.3P_t$; but the feed component (P_a) depends to a greater extent upon the various cutting conditions and may vary between $0.15P_t$ and $0.50P_t$.

The torque on the workpiece (for a workpiece of r in radius) becomes:

$$T = (P_t)\ (r) \tag{4.2}$$

This torque is applied by driving gear(s) on the spindle. Driving torque is equal to the torque on the workpiece in magnitude, if the efficiency of the bearings is assumed to be 100 %.

4.3.2 Driving Forces

The spindle of a lathe is driven by gear(s). In this study, gears are assumed to be spur gears and the following analysis is based on this specification.

It is possible to resolve a driving force, on a spur gear, into two main components as in Figure 4.2 :

1- Tangential component (Q_t) is tangent to the pitch circle of the gear. It is often called as the transmitted load, since it is really the useful component. The applied torque and the transmitted load are to be related by the equation:

$$T = (Q_t)\ (D_g /2) \tag{4.3}$$

where, D_g is the pitch circle diameter of the gear on the spindle.

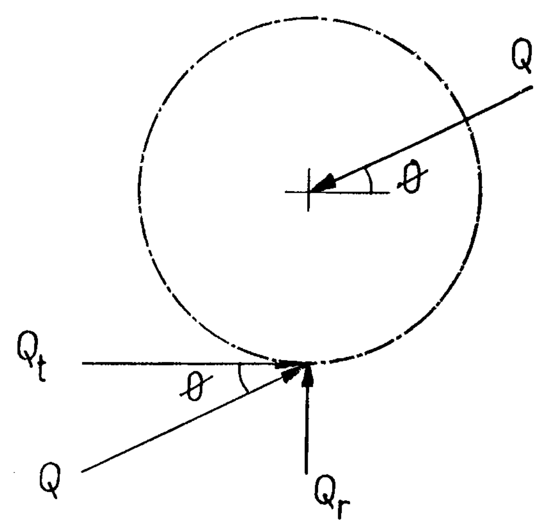


Fig.4.2. The Force Components on Spur Gears.

2- The radial component (Q_r) radially acts on the gear and it does not transmit power.

Since, the reactions between the mating teeth occur along the pressure line, the driving force component may be computed by the equations below:

$$Q_t = Q\ (\cos \phi) \qquad (4.4)$$

$$Q_r = Q\ (\sin \phi) \qquad (4.5)$$

and these components may be summed vectorially to give the total radial reaction (Q) on the spindle:

$$Q = \left[(Q_t)^2 + (Q_r)^2 \right]^{1/2} \qquad (4.6)$$

In the above formulae, ϕ is the pressure angle and it usually has values of 20° or 25°. In this study a 20° spur gear is used.

4.4. DESIGN FOR STIFFNESS

4.4.1. Stiffness

The question of stiffness is often more important in the design of metal cutting machine tools than that of load carrying capacity, because the stresses which correspond to the permissible deformations are generally much less in value than those permissible for the various materials.

Stiffness is usually defined as:

$$\text{Stiffness} = \frac{\text{Load}}{\text{deflection}}$$

The term "stiffness" has to be considered from the following points of view:

a) Static stiffness against deflection under loads which are considered as static.

b) Dynamic stiffness (rigidity), i.e. behaviour during vibrations under pulsating and inertia forces.

In this study, the main emphasis is given to the static stiffness.

Under the application of operating forces (cutting and driving forces) spindles exhibit deflections in radial and axial directions. This requires the evaluation of radial and axial stiffnesses of the spindles.

Radial stiffness of the spindle K_{sr}, may be defined as the ratio of the force P exerted on the spindle nose to the displacement at the spindle nose Y_T. That is:

$$K_{sr} = \frac{P}{Y_T} \tag{4.7}$$

Axial stiffness, however, is defined as the ratio of P_a to the axial deflection of the spindle nose. It is expressed as:

$$K_{sa} = \frac{P_a}{Y_{sa}} \qquad (4.8)$$

In the above expressions, P and P_a are specified. Hence, the deflection must be evaluated to determine stiffnesses in radial and axial directions.

4.4.2. Deflection Analysis in Radial Direction:

4.4.2.1. General Consideration and Setting up the Model

During the operation of the spindle the displacement at the spindle nose is caused not only by force P, but also by tilting moment M_t, produced by axial component of cutting force and exerted on the spindle nose, and by driving force Q, produced by driving elements and exerted on a certain point of the spindle. Since the components of the cutting force P vary due to the cutting conditions, P, Q and M_t do not lie in the same plane. The total deflection at nose is a vector quantity and equals to the vector sum of:

$$\overrightarrow{Y_s} = \overrightarrow{Y_{sp}} + \overrightarrow{Y_{sq}} + \overrightarrow{Y_{sm}} \qquad (4.9)$$

However the total deflection at the nose can be made as small as possible by enforcing Y_{sp}, Y_{sq} and Y_{sm} partly offset each other in sensitive direction, i.e. The direction which exerts a decisive influence on machining accuracy.

When only a force P is exerted on the spindle nose, the radial deflection created at the spindle nose is equal to the superposition of deflections due to the span, overhang, front and rear bearings separately. This is shown in Figure 4.3.

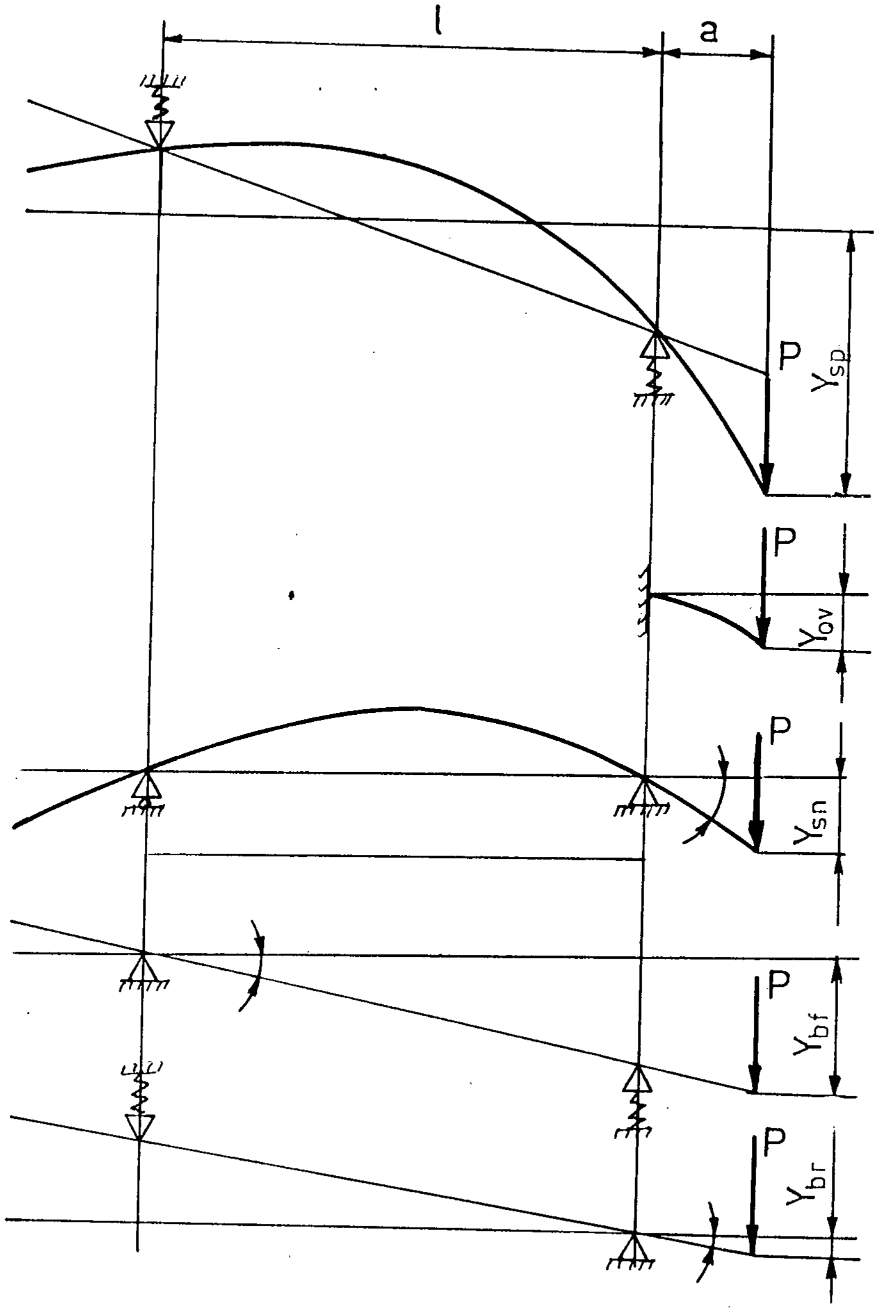


Fig.4.3. Spindle Deflection, when only P is Exerted on the Spindle Nose.

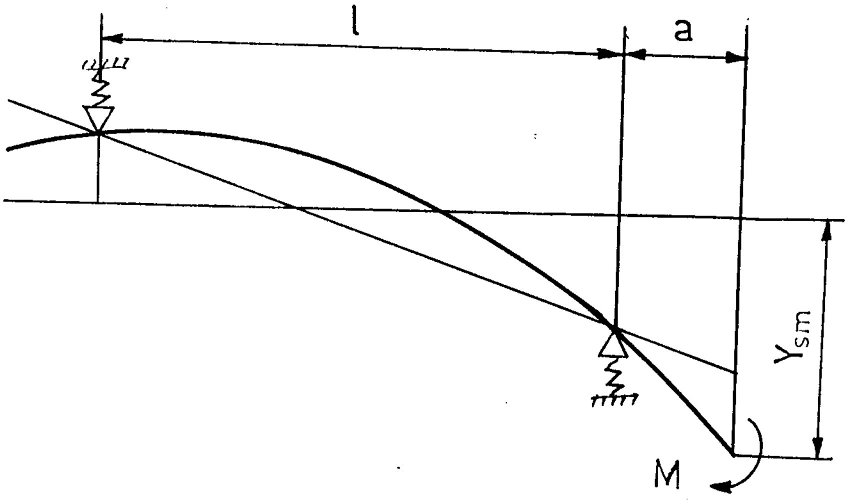


Fig.4.4. Spindle Deflection, when only M_t is Exerted on the Spindle Nose.

Figure 4.4 shows the deflection of spindle on which, only M_t acts.

The driving force Q may appear on the span, on the overhang or on the protruding tail.

Figure 4.5 shows the situation in which only the force Q acts on the overhang. It is shown that the deflections due to the overhang and two bearings are all in the same direction to pile up.

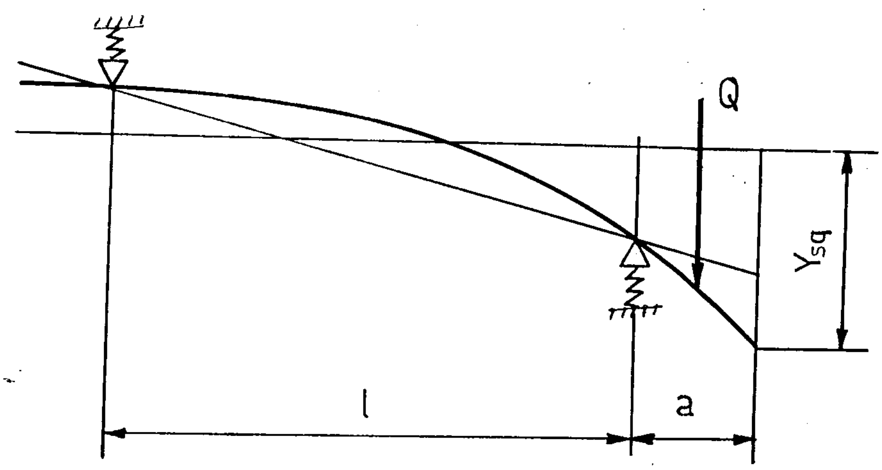


Fig.4.5. Spindle Deflection, when only Q is Exerted on the Overhang.

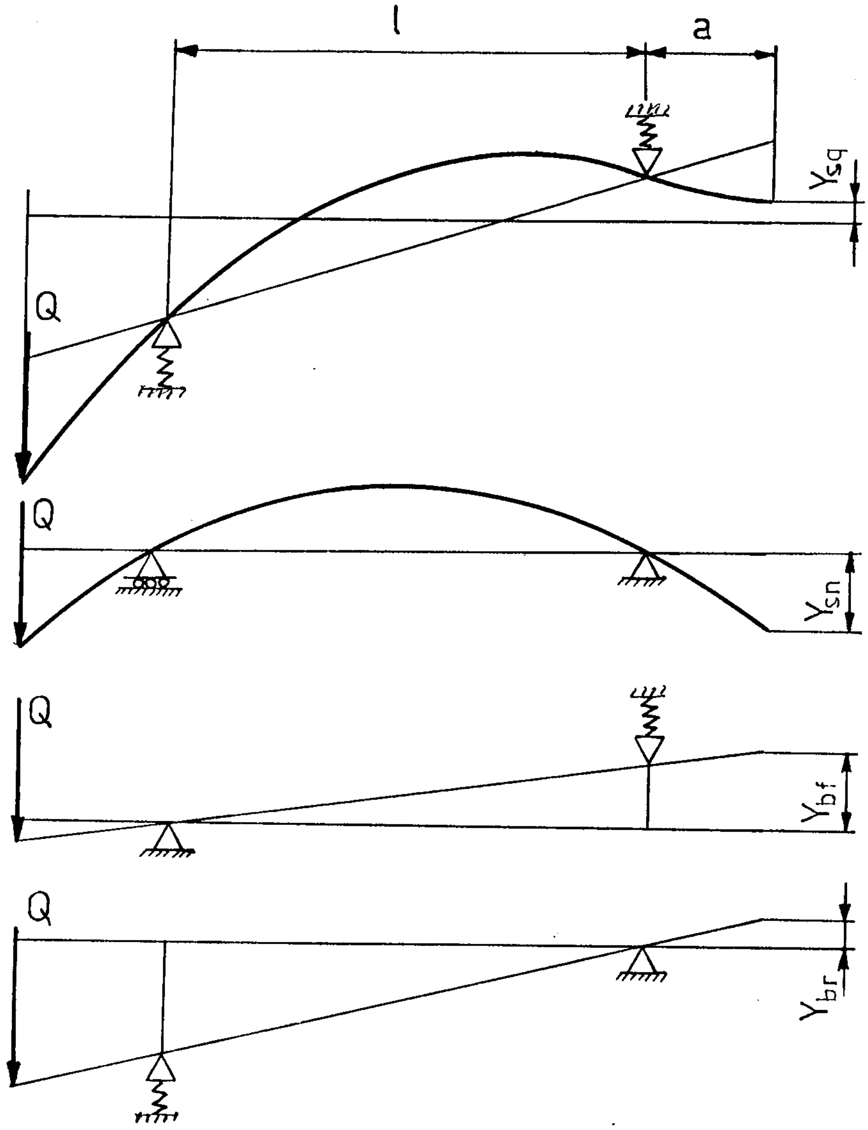


Fig.4.6. Spindle Deflection, when only Q is Exerted on the Protruding Tail.

If driving force Q acts on the protruding tail as in Figure 4.6, the radial deflection at the spindle-nose produced by the deflection of bearings are opposite in direction and that they can be offset by each other.

Figure 4.7 shows the situation in which driving force Q, acts on the span. The deflection at the spindle nose due to the deflection of the spindle and the rear bearing and that due to the deflection of the front bearing are opposite in direction, so they can be offset by each other. SHUZI (34) states that, under the action of force P, if span is less than the optimum

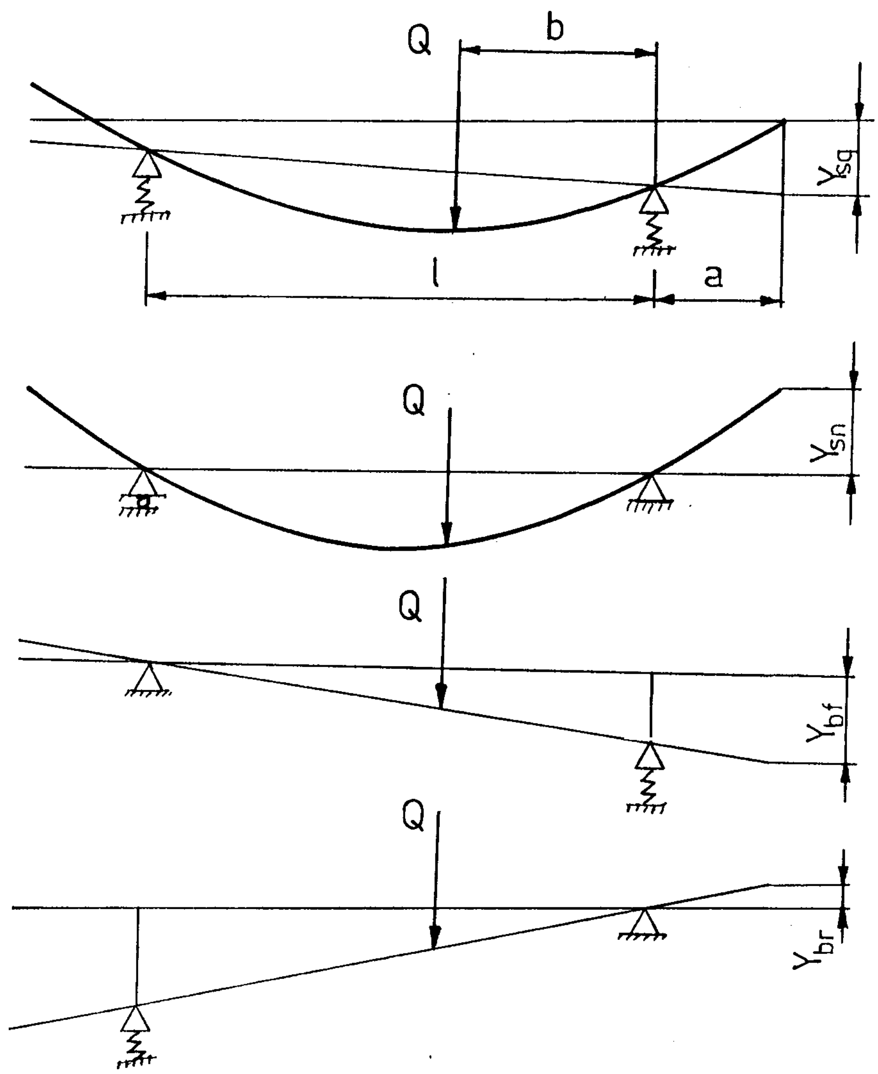


Fig.4.7. Spindle Deflection, when only Q is Exerted on the Span.

span (optimum span : the distance between bearings when deflection at nose is minimum) the tail will tilt up, but if it is greater than the optimum span, spindle axis will intersect the original axis. This point is called as node j, and if force Q is acted an the span, in front of this point as shown in Figure 4.8, then the nose deflections due to P and Q are in opposite sense to lessen the total radial deflection.

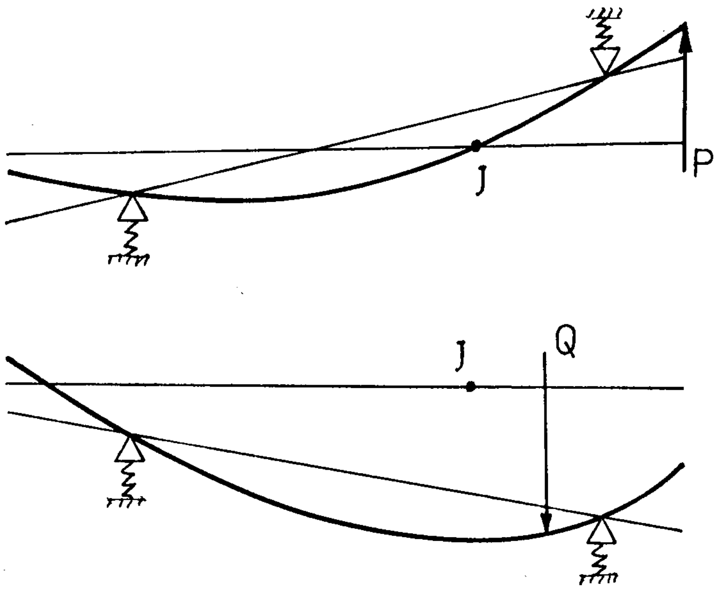


Fig.4.8. Fig.4.8. Definition of "node j"

As briefly discussed in the previous chapter, the model for the spindles is considered to be multi-step shaft supported at two points as shown in Figure (4.9)

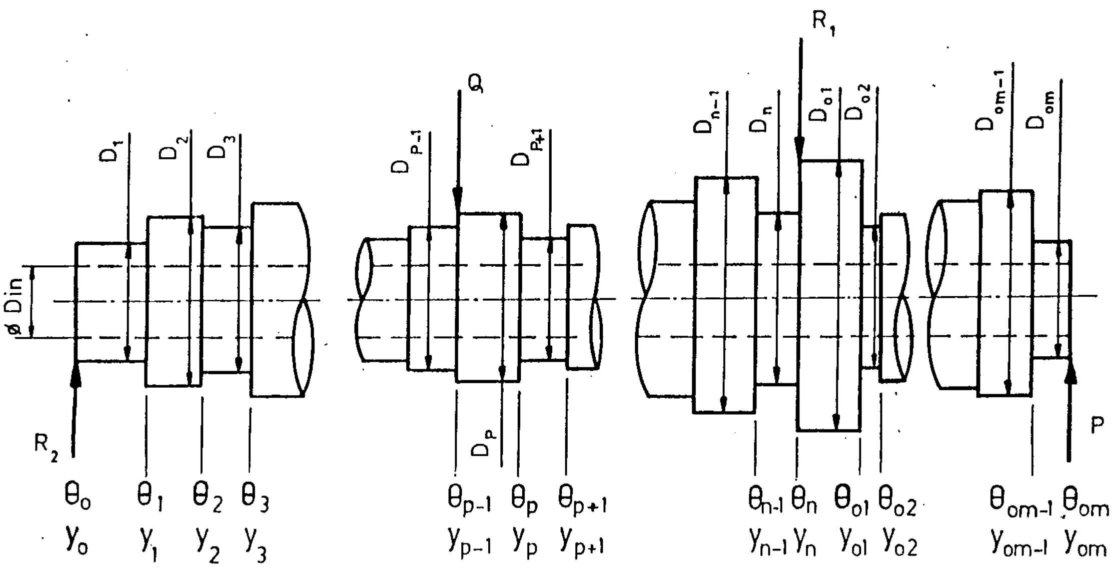


Fig.4.9. The Model for the Deflection Analysis of Spindles.

In radial deflection analysis, the following assumptions are made:

- Axial component of the cutting force is carried by the front bearings.
- Variations in the cutting force are neglected, i.e., cutting force remains constant.
- Driving gear is mounted on span in such a way that, the driving resultant radial load Q, produced by gear is in opposite direction, in the same plane as illustrated in Figure 4.10.

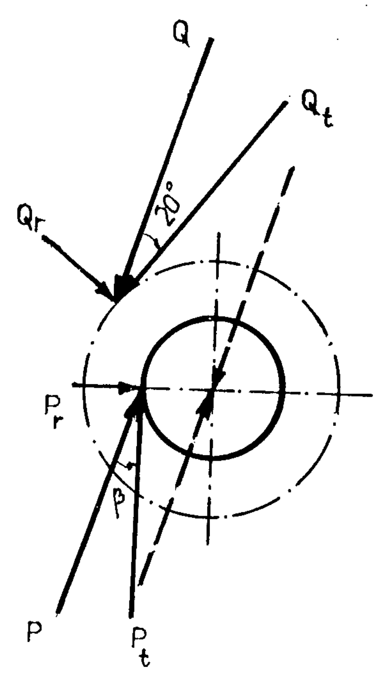


Fig.4.10. Assumption to Solve the Forces in a Plane.

In addition to these, the driving force reacts on the shoulder of the segment on which driving gear lies. It is proposed only for the deflection calculations to generalise the deflection formula and it is thought that, it does not cost to the designer very much unless the length of the segment is too long. Besides, it is possible to divide this segment into three parts with equal outer diameters to obtain the force segment at the middle with fairly small length. By applying the superposition technique to the model, the total radial deflection at the nose of the spindle is obtained as the sum of deflections due to spindle itself and the deflections due to the bearings:

$$Y_T = Y_s + Y_b \tag{4.10}$$

The deflection due to the spindle itself is treated in two parts :

1- Deflection due to the span : The span of the spindle may be assumed as a simply supported beam. The effect of the overhang on this part is included as a moment which is equal to the sum of the tilting moment at the tip and the moment created by the force P through the overhang a_o.

2- Deflection due to the overhang : The overhang is assumed as a cantilever beam. The deflection of this part is directly added on Y_{sp} to obtain the total deflection at the nose.

Hence, the nose radial deflection due to the spindle itself is equal to :

$$Y_s = Y_{sn} + Y_{ov} \tag{4.11}$$

The nose radial deflection due to the bearings consists of the effects of both bearings.

The nose deflection, due to the deflection at front bearings is :

$$Y_{bf} = \left(\frac{l + a}{l}\right) \delta_{fb} \tag{4.12}$$

and that due to the rear bearing is :

$$Y_{br} = \left(\frac{a}{l} \right) \delta_{rb} \tag{4.13}$$

Their sum gives the total effect of bearings on the spindle nose radial deflection :

$$Y_b = Y_{bf} + Y_{br} \tag{4.14}$$

4.4.2.2. Calculation of the Radial Deflection

Since the model is a multi-step shaft, evaluation of Y_{sn} and Y_{ov} involves complicated calculations. For the deriation of the formulae for these two terms the classic method which is described below is adapted.

When a shaft is subjected to a bending moment, the relationship between the radius of curvature and the moment is expressed by a well known formula as :

$$\frac{1}{\rho} = \frac{M}{EI} \tag{4.15}$$

where, ρ is the radius of curvature of the bent curve and it is given as :

$$\frac{1}{\rho} = \frac{\dfrac{d^2y}{dx^2}}{\left[1 + \left(\dfrac{dy}{dx} \right)^2 \right]^{3/2}} \tag{4.16}$$

where, y is the deflection of the beam at any point x along the beam. The slope of a beam is defined as :

$$\theta = \frac{dy}{dx} \qquad (4.17)$$

and in general the slope is very small, and denominator of Equation (4.16) can be taken as unity.

Equation (4.15) can then be written as :

$$\frac{M}{EI} = \frac{d^2y}{dx^2} \qquad (4.18)$$

It is possible to obtain slope and deflection relations by taking integral of Equation (4.18) twice. They become:

$$\theta = \frac{dy}{dx} + C_1 \qquad (4.19)$$

$$y = f (x) + C_2 \qquad (4.20)$$

where C_1 and C_2 are the boundary constants which can be obtained by entering the ends of the beam :

Evaluation of Y_{sn}:

To adopt the above for the evaluation of deflections for our model, the general method suggested by POPOV (36) is used:

He has introduced a method to obtain the slope and deflection equations of shafts with changing cross-section. This method consists of selecting an origin at one end of the beam and carrying out successive integrations until expressions for slope and deflection are obtained for the first segment. Due to the continuity conditions, these become the initial constants in the equations carried out for the next segment. This process is repeated until the far end of the beam is reached, then the boundary conditions of shaft are imposed to determine the remaining unknown constants. A new origin is used at the beginning of each segment. Any change on force or on inertias should result a new segment.

Application of this method to obtain the deflection equation of a shaft with in span segments in different lengths leads to the equation below :

$$
\begin{aligned}
y_n = {} & \frac{R_2}{6E}\left(\sum_{i=1}^{p-1} \frac{l_i^3}{I_i}\right) + \frac{R_2 - Q}{6E}\left(\sum_{i=p}^{n} \frac{l_i^3}{I_i}\right) + \frac{R_2}{2E}\left[\sum_{i=2}^{p} \frac{\left(\sum_{k=1}^{i-1} l_k\right) l_i^2}{I_i}\right] \\
& + \frac{R_2}{2E}\left[\sum_{i=p+1}^{n} \frac{\left(\sum_{k=1}^{p-1} l_k\right) l_i^2}{I_i}\right] + \frac{R_2 - Q}{2E}\left[\sum_{i=p+1}^{n} \frac{\left(\sum_{k=p}^{i-1} l_k\right) l_i^2}{I_i}\right] \\
& + \frac{R_2}{2E}\left[\sum_{i=2}^{p} l_i\left(\sum_{k=1}^{i-1} \frac{l_k^2}{I_k}\right)\right] + \frac{R_2}{2E}\left[\sum_{i=p+1}^{n} l_i\left(\sum_{k=1}^{p-1} \frac{l_k^2}{I_k}\right)\right] \qquad (4.21) \\
& + \frac{R_2 - Q}{2E}\left[\sum_{i=p+1}^{n} l_i\left(\sum_{k=p}^{i-1} \frac{l_k^2}{I_k}\right)\right] + \frac{R_2}{E}\sum_{i=3}^{p+1} l_i\left[\sum_{j=2}^{i-1} \frac{\left(\sum_{k=1}^{j-1} l_k\right) l_j}{I_j}\right] \\
& + \frac{R_2}{E}\sum_{i=p+2}^{n} l_i\left[\sum_{j=2}^{p-1} \frac{\left(\sum_{k=1}^{j-1} l_k\right) l_j}{I_j}\right] \\
& + \frac{R_2 - Q}{E}\sum_{i=p+2}^{n} l_i\left[\sum_{j=p+1}^{i-1} \frac{\left(\sum_{k=p}^{j-1} l_k\right) l_j}{I_j}\right] + \theta_o\left(\sum_{i=1}^{n} l_i\right)
\end{aligned}
$$

A deflection equation for an i th step produced by this approach consists of the terms of deflection of itself; terms of deflection resulting from the inclination of the segment just before, and the total deflection of the previous segments.

For a simply supported shaft the boundary conditions are :

$$C_1 = \theta_o \quad \text{at } x = 0$$

$$C_2 = 0 \quad \text{at } x = 0 \text{ and } x = 1$$

By applying the boundary conditions to the Equation (4.31), θ_o is calculated.

The inclination angle at the end of the n'th segment may be calculated from the derived formula below :

$$\theta_n = \frac{R_2}{2E}\left(\sum_{i=1}^{p-1} \frac{l_i^2}{I_i}\right) + \frac{R_2 - Q}{2E}\left(\sum_{i=p}^{n} \frac{l_i^2}{I_i}\right) + \frac{R_2}{E}\left[\sum_{i=2}^{p} \frac{\left(\sum_{k=1}^{i-1} l_k\right) l_i}{I_i}\right]$$

$$+ \frac{R_2}{E}\left[\sum_{i=p+1}^{n} \frac{\left(\sum_{k=1}^{p-1} l_k\right) l_i}{I_i}\right] + \frac{R_2 - Q}{E}\left[\sum_{i=p+1}^{n} \frac{\left(\sum_{k=p}^{i-1} l_k\right) l_i}{I_i}\right] + \theta_o \qquad (4.22)$$

P Previously found θ_o is inserted into the equation (4.22) to complete the calculation of θ_n.

Y_{sn} can be calculated by assuming the overhang is completely rigid. Then :

$$Y_{sn} = a_o \times \theta_n \qquad (4.23)$$

Evaluation of Y_{ov} :

For the case of overhang, this part may be assumed as a cantilever beam, so that its end conditions become:

$C_1 = 0$ at $x = 0$, since $\theta_o = 0$ for contililevers.

$C_2 = 0$ at $x = 0$

The deflection of overhang part may be derived as :

$$y_{om} = -\frac{P}{6E}\left(\sum_{i=1}^{m}\frac{a_i^3}{I_{oi}}\right) - \frac{P}{2E}\left[\sum_{i=2}^{m}\frac{\left(\sum_{k=1}^{i-1}a_k\right)a_i^2}{I_{oi}}\right] \tag{4.24}$$

$$+\frac{M + PA}{2E}\left(\sum_{i=1}^{m}\frac{a_i^2}{I_{oi}}\right) - \frac{P}{2E}\left[\sum_{i=2}^{m}a_i\left(\sum_{k=1}^{i-1}\frac{a_k^2}{I_{ok}}\right)\right]$$

$$-\frac{P}{E}\sum_{i=3}^{m}a_i\left[\sum_{j=2}^{i-1}\frac{\left(\sum_{k=1}^{j-1}a_k\right)a_j}{I_{oj}}\right] + \frac{M + PA}{E}\left[\sum_{i=2}^{m}a_i\left(\sum_{k-1}^{i-1}\frac{a_k}{I_k}\right)\right]$$

Equation (4.24) consist of the similar terms with the ones in span part and

$$Y_{ov} = Y_{om} \tag{4.25}$$

In the above equations, R_2 is expressed as :

$$R_2 = \frac{\left(\sum_{i=1}^{m}a_i\right)P + M_t + Q\left(\sum_{i=p}^{n}l_i\right)}{\left(\sum_{i=1}^{n}l_i\right)}$$

4.4.3. Deflection Analysis in Axial Direction :

Axial deflection (or stiffness) of spindles is also a summation of deflections through the spindle itself, spacers and especially axial deflections of bearing.

$$Y_{Ta} = Y_{sa} + Y_{ba} \tag{4.26}$$

The axial deflection through the spindle itself and the other construction elements like spacers can be calculated by well known compressive deformation formula :

$$Y_{sa} = \sum_{i} \frac{P_a \, a_{oi}}{E_i \, A_i} \tag{4.27}$$

4.5. DESIGN FOR STRENGTH

During iterative design of spindles, the designer may hesitate to specify the initial outer diameter of the spindle. For its simplicity in application, ASME Code formula is used to aid the designer in his first guess on the outer diameter.

The main design criterion of spindles is deflection. So that, a design satisfying the required minimum deflection condition will have ample strength. But a strength check for infinite life is performed here, for the sake of completeness.

4.5.1. ASME Code Formula

For a hollow shaft with internal diameter ratio:

$$r_{io} = \frac{D_i}{D_o} \tag{4.28}$$

ASME code formula is (37)

$$D_o^3 = \frac{16}{\pi \tau_d (1-r_{io}^4)} \left[(K_s \; T)^2 + \left(K_m \; M + \frac{\alpha F_a D_o (1+r_{io}^2)}{8} \right)^2 \right]^{1/2} \quad (4.29)$$

where, τ_d is design shear stress and it is equal to one of the below which one is smaller:

$$\tau_d = (0.3)\; S_y \quad (4.30)$$

or

$$\tau_d = (0.18)\; S_u$$

The design stress should be modified for keyways :

$$\tau_d = (0.75)\tau_d \quad (4.31)$$

K_s and K_m are shock and fatigue factors for torsion and bending moment respectively. They are tabulated in Table 4.1.

Table 4.1. - Values of K_m and K_s (Ref. 37).

Nature of Loading	K_s	K_m
Stationary Shafts		
(Bending Stress not reversed		
Gradually applied	1.0	1.0
Suddenly applied	1.5 to 2.0	1.5 to 2.0
Rotating shafts		
(Bending stress reversed)		
Gradually applied or steady	1.5	1.0
Sudenly applied, minor shocks	1.5 to 2.0	1.0 to 1.5
Sudenly applied, heavy shocks	2.0 to 3.0	1.5 to 3.0

For the running condition of spindles (suddenly applied, minor shocks) K_s and K_m is specified as 1.5 and 2.0 respectively. For circular cross-sections

$$\alpha = \frac{1}{1-\left[0.0044(4L/D)\right]} \quad \text{for } \frac{L}{D} < 28.75 \qquad (4.32)$$

$$\alpha = \frac{S_y}{\pi^2 nE}\left(\frac{4L}{D}\right)^2 \quad \text{for } \frac{L}{D} > 28.75 \qquad (4.33)$$

According to the proposed spindle model, the maximum moment occurs on the section of front bearings. And also considering the overhang of a spindle has relatively large diameter, the critical section is selected on the span and it is assumed that there exists no axial force.

4.5.2. Design By Soderberg Approach

The model for the strength program is shown in Figure 4.11.

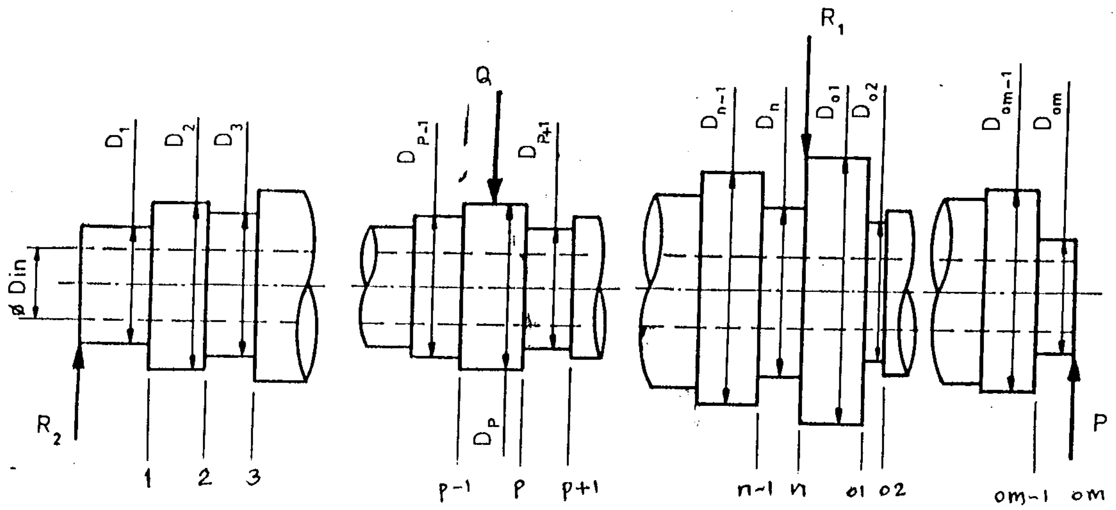


Fig.4.11. The Model for Strength Program.

According to the loading conditions in Figure there exist only reversed bending in the part of the spindle from rear bearing to the driving gear. The other part of the spindle is subjected to reversed bending and constant torsional shear stresses. They are equal to :

$$\sigma_{xa} = \frac{Mc}{I} \tag{4.34}$$

$$\tau_{xym} = \frac{Tr}{J} \tag{4.35}$$

They are calculated for each section and are displayed to choose the critical section by the designer. They are combined by Von-Mises stress formula, below :

$$\sigma'_m = \left[\sigma^2_{xm} + 3\tau_{xym}\right]^{1/2} \tag{4.36}$$

$$\sigma'_a = \left[\sigma^2_{xa} + 3\tau_{xya}\right]^{1/2} \tag{4.37}$$

For the case of spindles, Von-Mises stresses are reduced to :

$$\sigma'_m = \sqrt{3}\,\tau_{xym} \tag{4.38}$$

$$\sigma'_a = \sigma_{xa} \tag{4.39}$$

With the connection of Mises-Hencky theory, the Soderberg line on the fatigue diagram gives the formula for factor of safety :

$$FS = \frac{1}{\dfrac{\sigma'_a}{Se} + \dfrac{\sigma'_m}{S_y}} \tag{4.40}$$

where, Se is called as the modified endurance limit and equal to :

$$Se = Se' \, (k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e) \qquad (4.41)$$

where, Se' is the endurance limit of the rotating-beam specimens.

$$Se' = 0.5 \; S_{ut} \quad \text{if } S_{ut} < 1400 \text{ MPa} \qquad (4.42)$$

$$Se' = 700 \text{ MPa} \quad \text{if } S_{ut} \geqslant 1400 \text{ MPa}$$

k_a, k_b, k_c, k_d, k_e are modification factors :

k_a is the surface factor and in the case of spindles for ground surface : $k_a = 0.89$

k_b is the size factor. For bending and torsion it should be selected as :

$$k_b = 1 \qquad \text{for } d \leqslant 8 \text{ mm}$$

$$k_b = 1.189 \; d^{-0.097} \qquad \text{for } 8 \text{ mm} < d \leqslant 250 \text{ mm}$$

It is calculated in the program automatically. k_c is reliability factor and may be selected from (Ref. 35). It is enough to give percent reliability in the CAD package.

k_d is temperature factor and for the case of spindle design :

$$k_d = 1.0$$

k_e is used to compensate the effect of stress concentration. It is calculated as :

$$k_e = \frac{1}{K_f}$$

where, K_f is called as fatigue strength reduction factor and it is equal to :

$$K_f = 1 + q(K_t - 1) \tag{4.44}$$

where, q is notch sensitivity factor and K_t is stress concentration factor. They may be calculated by means of charts which exist in machine design textbooks. K_t and q values should be supplied for the Spindle Design Program.

4.6. CRITICAL SPEED

The first mode of natural frequency of a shaft is usually called as the critical speed of that shaft.

Rayleigh-Ritz equation is a simple and highly useful expression for determining the critical speed of shafts. The equation is written by static forces on the shaft, which are generally the weights of the components on the shaft and the weight of the shaft itself, and the deflections, they caused. It does not give the exact value because, the curve of statical deflections is not exactly proportional to the dynamic deflection curve used in the calculation of natural frequency of rotating bodies. However, the result obtained by the equation is only a few percent higher than the true natural frequency (38). The Rayleigh-Ritz equation for the first critical speed of a rotating shaft is :

$$N_c = \frac{30}{\pi} \left[\frac{g \sum_i W_i y_i}{\sum_i W_i y_i^2} \right]^{1/2} \tag{4.45}$$

where, N_c : The first critial speed of the shaft (vpm)

g : Gravitational constant (m/s^2)

W_i : Weight of the i'th element (N)

y_i : Static deflection of the i'th element (mm)

To include shaft mass in the calculations, the shaft may be divided into several parts, each of them is treated as an additional mass.

A spindle of a lathe, being a rotating shaft, should be controlled not to reach its critical speed.

In this analysis, the model is considered as divided into four sections, as shown in Figure 4.12. The respective weights of which are :

$$W_I = \sum_{i=1}^{p-1} W_i$$

$$W_{II} = W_p \tag{4.46}$$

$$W_{III} = \sum_{i=p+1}^{n} W_i$$

$$W_{IV} = \sum_{j=1}^{m} W_j$$

The weight forces are considered to be acting at the gravity centers of sections 1,2,3 and 4.

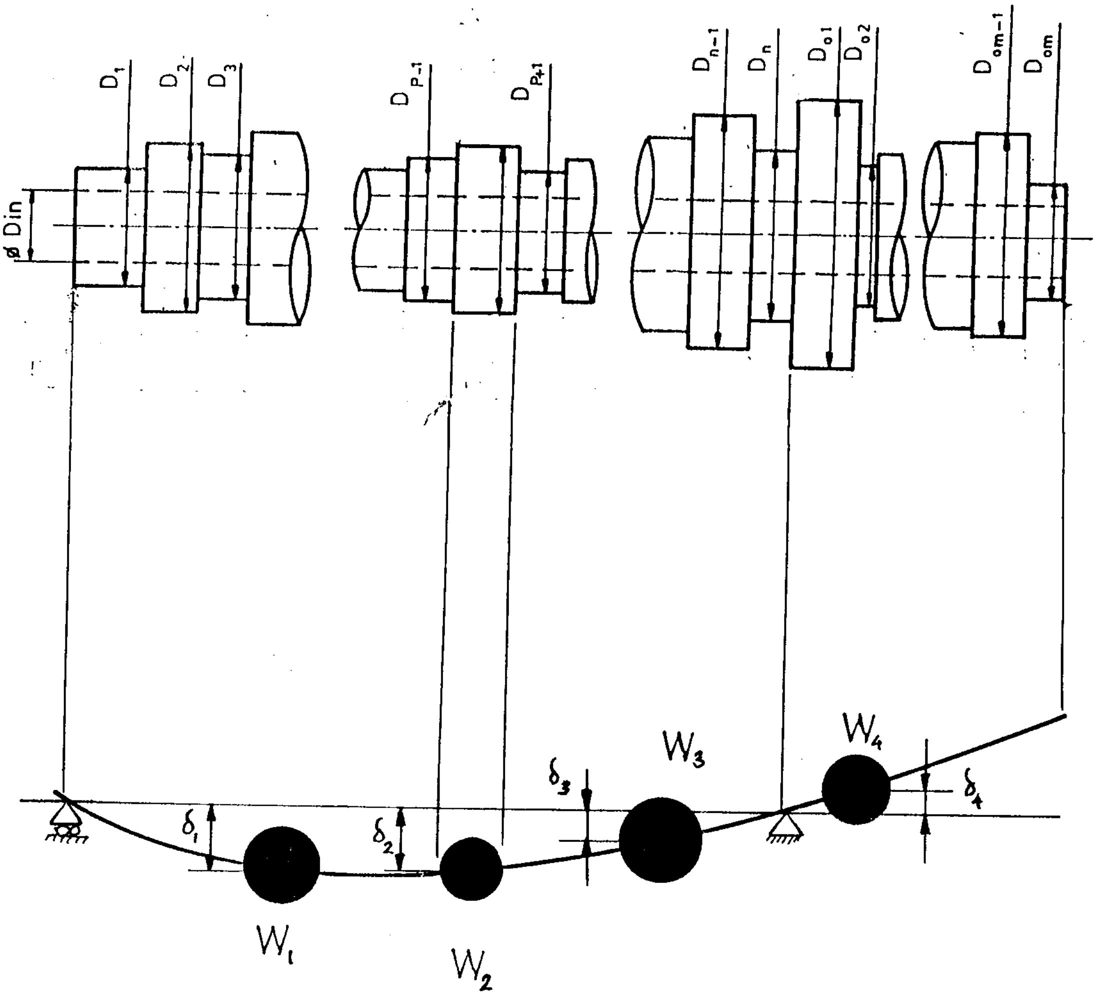


Fig.4.12. The Model for Critical Speed Calculation.

CHAPTER 5

THE APPLICATION OF CAD TO SPINDLE DESIGN

5.1. INTRODUCTION

This chapter is to illustrate the systematic approach developed in this thesis on the design of the spindle of lathes.

The general view of the CAD package which automates the design approach is presented in Section 5.2. The use of the package is illustrated with an example in the next section.

5.2. GENERAL VIEW OF THE CAD PACKAGE

The computer Aided Design package consists of the individual programs which performs the calculations of the deflection, the strength and the critical speed. The individual programs are brought into action by giving proper answers to the questions asked by the program at different stage of the design procedure.

The package works on an interactive basis. The flow-chart is given in Appendix 5.1. It enables the designer to obtain an optimum design searching through various alternatives providing that the essential data is input. The flow of the program for a specifie run is presented in Appendix 5.2.

At the first entry to CAD package, a message on how the input data is to be fed, is typed out on the terminal.

The input data to the program contains radius of the work piece, tangential component of cutting force, pitch diameter of the gear, modulus of elasticity, ultimate and yield strength of the spindle material, number of steps on span and overhang and the number of the step on which the driving gear is mounted.

Once the data is set up, the following question is typed out on the terminal :

WHICH NOW ?

The answer BGN bring the action to the beginning of the program to change some of the input data. The answer SPL brings the Spindle Analysis program into action. To terminate the program EXT command is given.

If the answer is SPL, the question of whether the workpiece on the spindle or not is asked. After giving the proper answer, the following is typed out on the terminal :

WHAT NOW ?

At this point, the user has a lot of options. As illustrated in Appendix 5.2, by the proper answers to the question above new set of data can be given, previous data can be read from the disc area, any of the parameters can be changed and by the command CHK the data can be tabulated on the terminal. The data includes length and outer diameters of the span and overhang segments, inner diameter of the spindle, radial and axial stiffnesses of front bearing and radial stiffness of the rear bearing. After setting and checking up the data, with the answer RUN, the reactions at the supporting points are calculated and then, the following question is seen on the terminal.

The input data to the program contains radius of the work piece, tangential component of cutting force, pitch diameter of the gear, modulus of elasticity, ultimate and yield strength of the spindle material, number of steps on span and overhang and the namber of step which the driving gear on it.

Once the data is set up, the following question is typed out on the terminal :

WHICH NOW ?

The answer BGN bring the action to the beginning of the program to change some of the input data. The answer SPL brings the Spindle Analysis program into action. To terminate the program EXT comment is given.

If the answer is SPL, the question of whether the workpiece on the spindle or not is asked. After giving the proper answer, the following is typed out on the terminal :

WHAT NOW ?

At this point, the user has a lot of options. As illustrated in Appendix 5.2, by the proper answers to the question above new set of data can be given, previous data can be read from the disc area, any of the parameters can be changed and by the command CHK the data can be tabulated on the terminal. The data includes length and outer diameters of the span and overhang segments, inner diameter of the spindle, radial and axial stiffnesses of front bearing and radial stiffness of the rear bearing. After setting and checking up the data, with the answer RUN, the reactions at the supporting points are calculated and then, the following question is seen on the terminal.

STRENGTH CHECK ? ANSWER (YES/NO)

The answer, YES, brings strength calculation program into action. Section modulus and T/J ratios are tabulated for the section of the span and the overhang. By choosing the maximums of those, and giving the values of reliability, notch sensitivity and stress concentration factors, factor of safety based on soderberg criteria is calculated, and the following is typed out on the terminal.

IS FACTOR OF SAFETY SATISFACTORY ? ANSWER (YES/NO)

The answer NO brings the program action to the beginning to try new values of the design parameters. If the factor of safety is satisfactory then strength calculation program is terminaled and deflection calculation program comes into action. This program calculates : Deflection gradients on front and rear bearings, radial deflection at nose due to span, overhang, and bearings and total radial and axial deflections on the nose together with gross radial and axial stiffnesses.

If the calculated deflections are not within the requirements of the spindle, the answer YES to the following question brings the program action to the point where the data of the spindle is set up.

ANOTHER TRIAL ? (YES/NO)

However, the answer terminates this program and the following is asked by the program.

PLOT THE RESULTS ? ANSWER (YES/NO)

The answer NO to this question makes the program to type out the following question :

DO YOU WANT TO CALCULATE CRITICAL SPEED ?

The answer YES to this question brings critical speed program into action.

After specifying the material of the spindle critical speed value is typed out and again the following question is seen on the terminal :

WHICH NOW ?

to set up a new data, to try another spindle or to exit from the program.

As noticed above, the calculations may be for a specific spindle where all design parameters are fixed or they may be for various conditions where one of the design parameter is varied while the rest are kept constant. For the lather case, the resülts are arranged in an array form to illustrate the variation of the nose deflection against that parameter using the graphic program. This option is brought by giving the answer YES to the question:

PLOT THE RESULTS ? (YES/NO)

5.3. COMPUTER AIDED DESIGN OF SPINDLES

5.3.1. Setting up the Example

A spindle with 6 steps in the span and 3 steps in overhang is chosen to illustrate the use of the CAD package. The configuration of this spindle is given in Figure 5.1.

The information in the Figure is fed to the program by the way described in the previous section and as illustrated in Appendix 5.2.

The effect of the design parameters such as span length, overhang length, outer and inner diameters of the spindle, and the position of the driving force on the total nose deflection are investigated in the following sections.

5.3.2. Span Length

Span length may be increased or decreased by increasing the lengths of span segments, while keeping the number of steps constant. Variation of radial deflection at nose due to the front and rear bearings and due to the spindle is given in Figures 5.2, 5.3 and 5.4 respectively. Figure 5.5 illustrates the effect of changes in span on the total radial deflection at nose. As it is expected, the increase in span, decreases the deflection at nose due to bearings but increases it due to spindle itself.

Total deflection at the nose exhibits a decrease at the beginning and then an increase while span length increases. Although smaller values of the span length have no practical value from manufacturing point of view, they have been included to illustrat this nature of the total deflection.

5.3.3. Outer and Inner Diameters

To illustrate the effect of the variation of the outer diameter on the nose deflection, outer diameters of each segment of the spindle are increased with the same amount.

Variation of the nose deflection with the outer diameter is illustrated in Figure 5.6. Increasing the outer diameter decreases the deflection in a considerable amount.

In the same manner, the variation of the nose deflection with the inner diameter is illustrated in Figure 5.7. Increasing the inner diameter, increases the nose deflection.

5.3.4. Overhang Length

Overhang length may be increased or decreased by increasing or decreasing the lengths of overhang segments while keeping the number of steps constant. Increasing the length increases the nose deflection as is illustrated in Figure 5.8. Since linearly dependent, axial deflection also increases with the increase in the overhang length. It is shown in Figure 5.9.

To include the effect of chuck and the workpiece, number of the overhang is increased by one. For this specific example, inclusion of chuck with the dimensions given in Figure 5.10 has increased the deflection about 3 times at the same point.

5.3.5. Position of the Driving Force

The program is capable of varying the position of the drive force to see its effect on the spindle nose deflection. For illustrative purposes 2 positions of the drive force has been investigated. Figures 5.11 and

5.12 show the results when the driving force is on the forth segment (i.e., 30 mm from the front bearing) and on the second segment (i.e., 70 mm from the front bearing), respectively. It can be seen from the Figures that, total radial deflection at nose increases with increase in the distance of the point of application of the driving force from the front bearing. It is noticed that there is no change in the characteristics of the deflection curve.

5.3.6. Simplification to One-Step Span and Overhang

A one-step equivalent of the 6-step spindle in Figure 5.1 is prepared for comparison purposes. It has equivalent (in mass moment of inertia) diameters, while keeping span and overhang lengths are constant. It is shown in Figure 5.13 with the results. The results show that total radial deflection for one-step model is 5 % higher than the six-step one. Percent error in critical speed reaches to 11 % and again it is higher then spindle model.

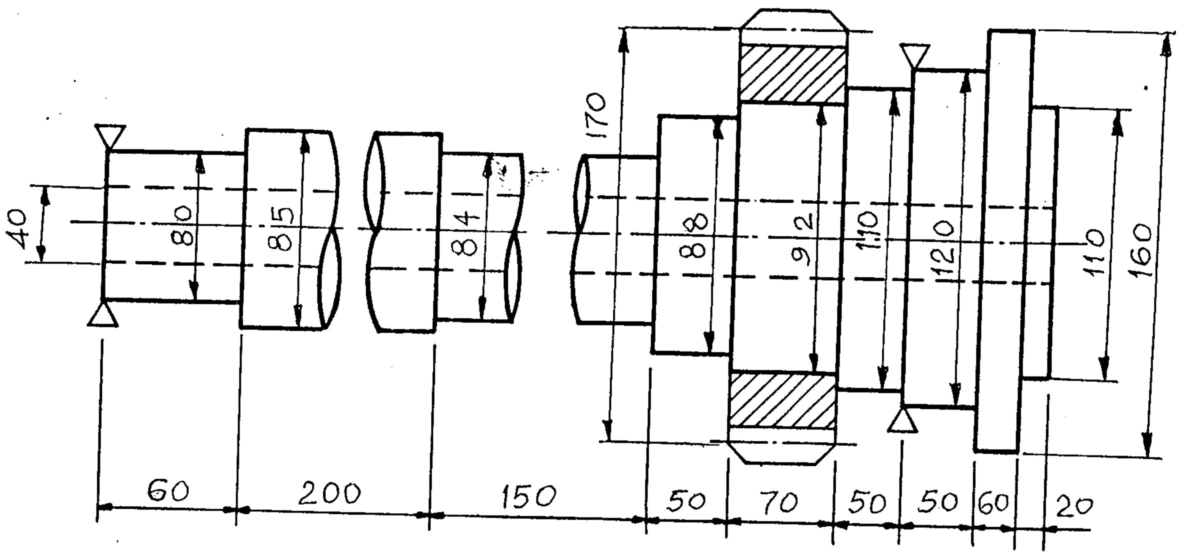
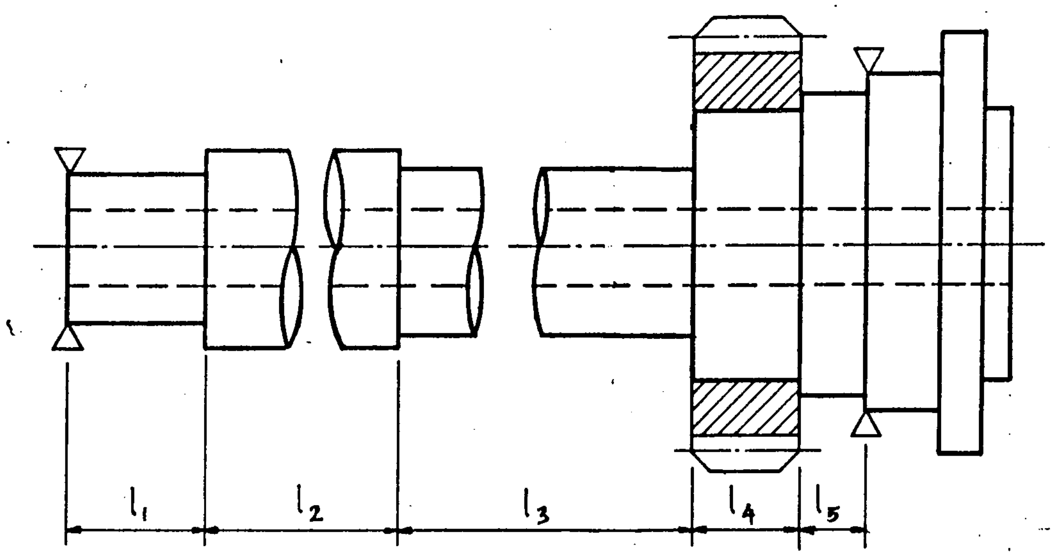


Fig.5.1. The Spindle in the Example



TRIAL NO	SPAN	DEF.FRONT BRG
1	X(1) = 30.00000(MM)	Y(1) = 0.01932(MM)
2	X(2) =100.00000(MM)	Y(2) = 0.01446(MM)
3	X(3) =110.00000(MM)	Y(3) = 0.01287(MM)
4	X(4) =120.00000(MM)	Y(4) = 0.01161(MM)
5	X(5) =150.00000(MM)	Y(5) = 0.00907(MM)
6	X(6) =200.00000(MM)	Y(6) = 0.00684(MM)
7	X(7) =250.00000(MM)	Y(7) = 0.00565(MM)
8	X(8) =350.00000(MM)	Y(8) = 0.00443(MM)

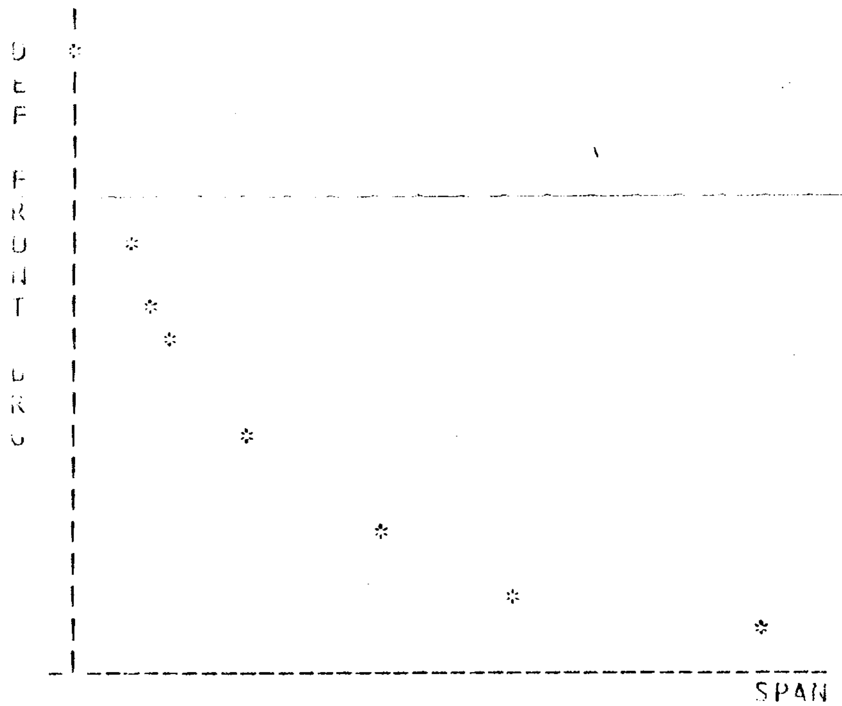
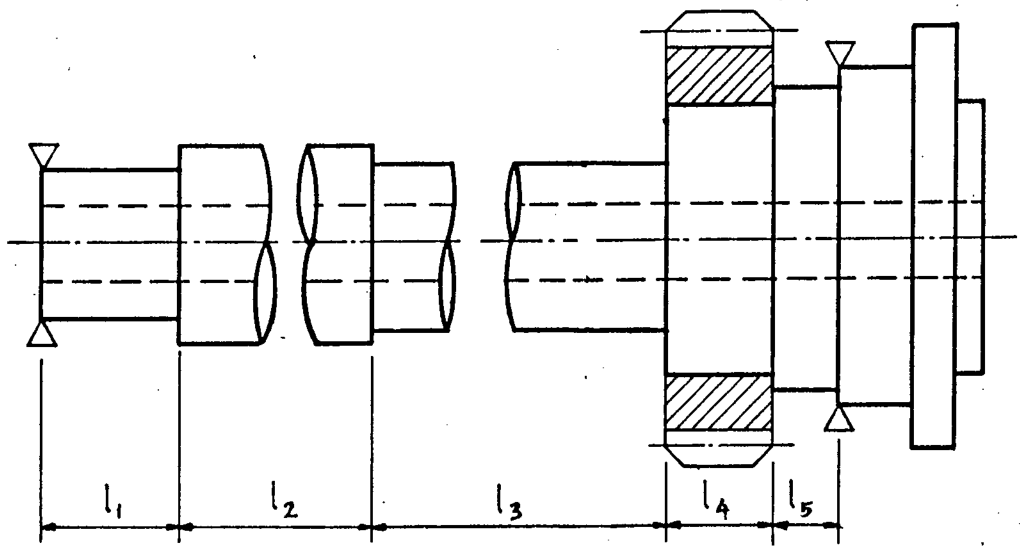


Fig.5.2. Variation of Radial Deflection at Nose due to the front Bearing.



TRIAL NO	SPAN	DEF.REAR BRG
1	X(1) = 80.00000(MM)	Y(1) = 0.00972(MM)
2	X(2) =100.00000(MM)	Y(2) = 0.00639(MM)
3	X(3) =110.00000(MM)	Y(3) = 0.00534(MM)
4	X(4) =120.00000(MM)	Y(4) = 0.00455(MM)
5	X(5) =150.00000(MM)	Y(5) = 0.00302(MM)
6	X(6) =200.00000(MM)	Y(6) = 0.00180(MM)
7	X(7) =250.00000(MM)	Y(7) = 0.00122(MM)
8	X(8) =350.00000(MM)	Y(8) = 0.00069(MM)

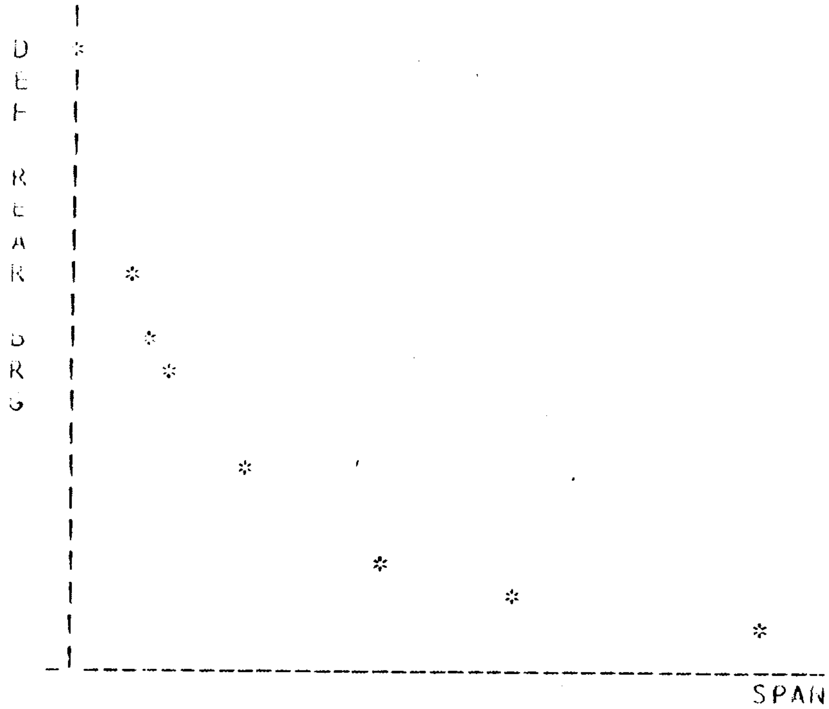
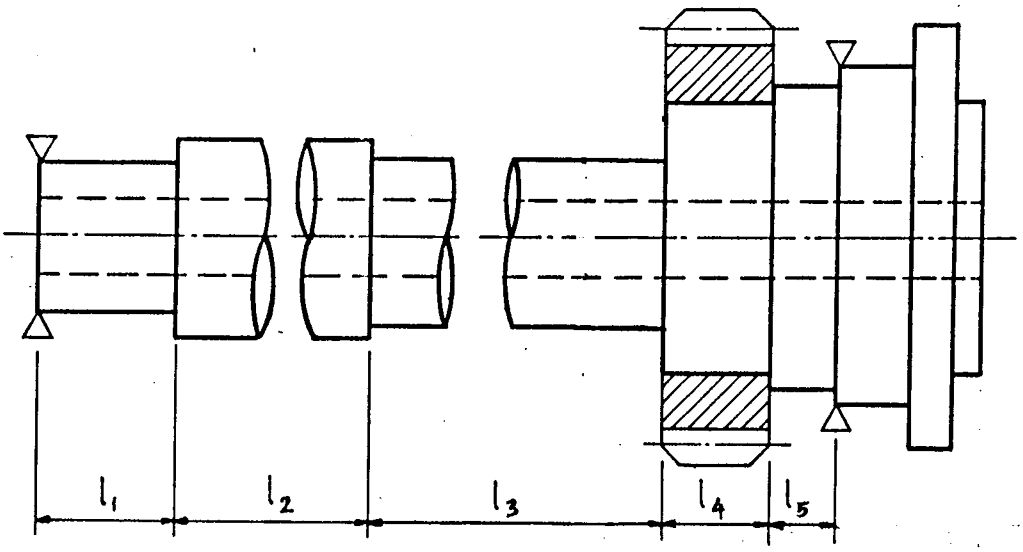


Fig.5.3. Variation of Radial Deflection at Nose due to the Rear Bearing.



TRIAL NO	SPAN	SPIN.R.DEF.
1	X(1) = 80.00000(MM)	Y(1) = 0.00710(MM)
2	X(2) =100.00000(MM)	Y(2) = 0.00808(MM)
4	X(3) =110.00000(MM)	Y(3) = 0.00857(MM)
3	X(4) =120.00000(MM)	Y(4) = 0.00907(MM)
5	X(5) =150.00000(MM)	Y(5) = 0.01060(MM)
6	X(6) =200.00000(MM)	Y(6) = 0.01327(MM)
7	X(7) =250.00000(MM)	Y(7) = 0.01609(MM)
8	X(8) =350.00000(MM)	Y(8) = 0.02219(MM)

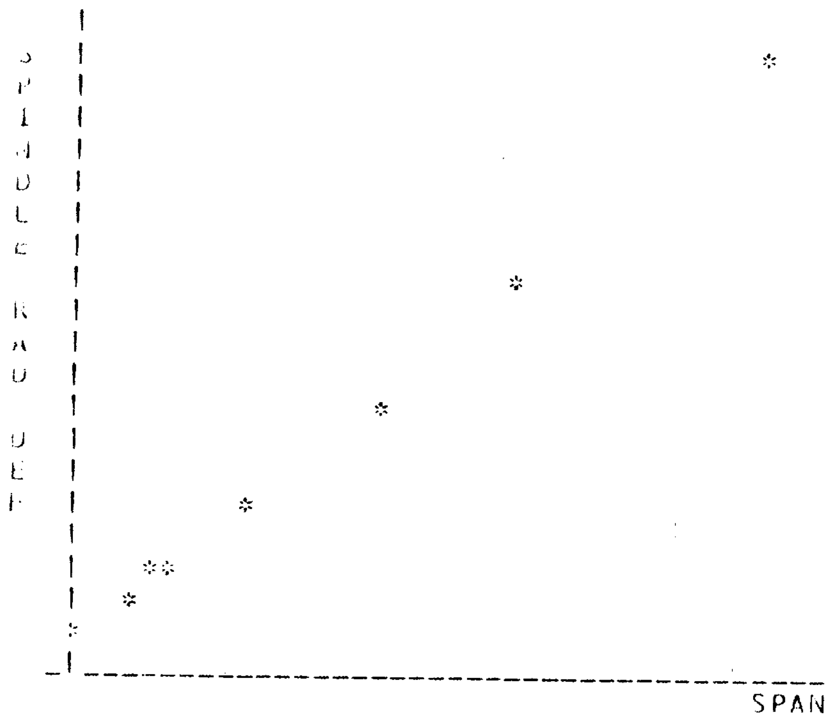
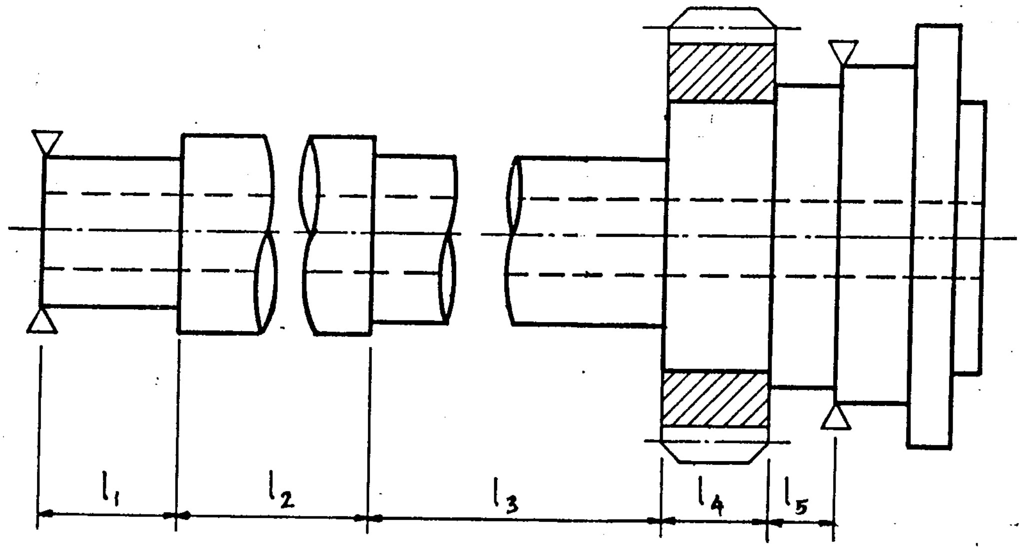


Fig.5.4. Variation of Radial Deflection at Nose due to the Spindle itself.



TRIAL NO	SPAN	TOT.DEF.
1	X(1) = 80.00000(MM)	Y(1) = 0.02194(MM)
2	X(2) =100.00000(MM)	Y(2) = 0.01277(MM)
4	X(3) =110.00000(MM)	Y(3) = 0.00964(MM)
3	X(4) =120.00000(MM)	Y(4) = 0.00709(MM)
5	X(5) =150.00000(MM)	Y(5) = 0.00148(MM)
6	X(6) =200.00000(MM)	Y(6) = 0.00463(MM)
7	X(7) =250.00000(MM)	Y(7) = 0.00922(MM)
8	X(8) =350.00000(MM)	Y(8) = 0.01707(MM)

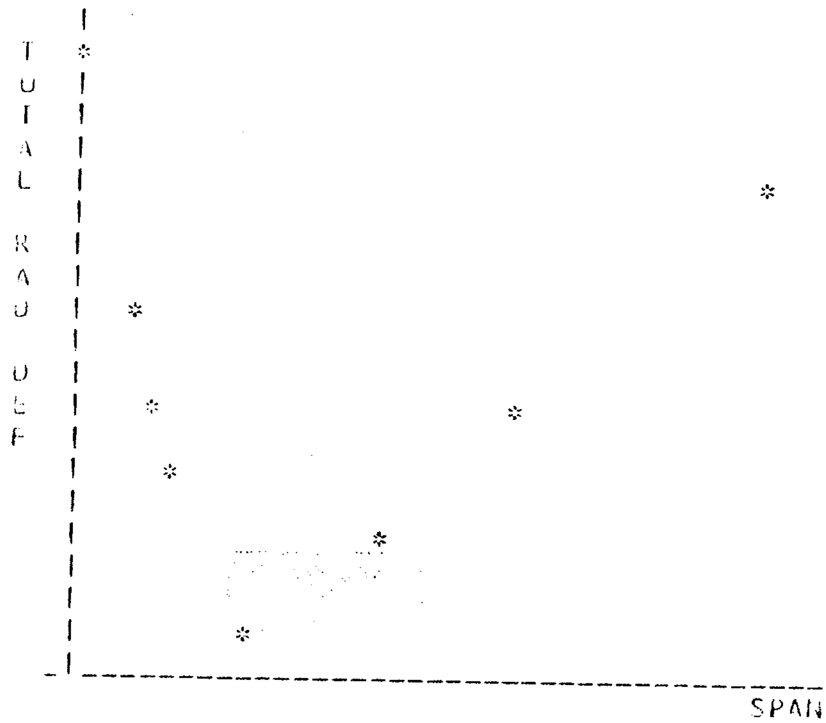
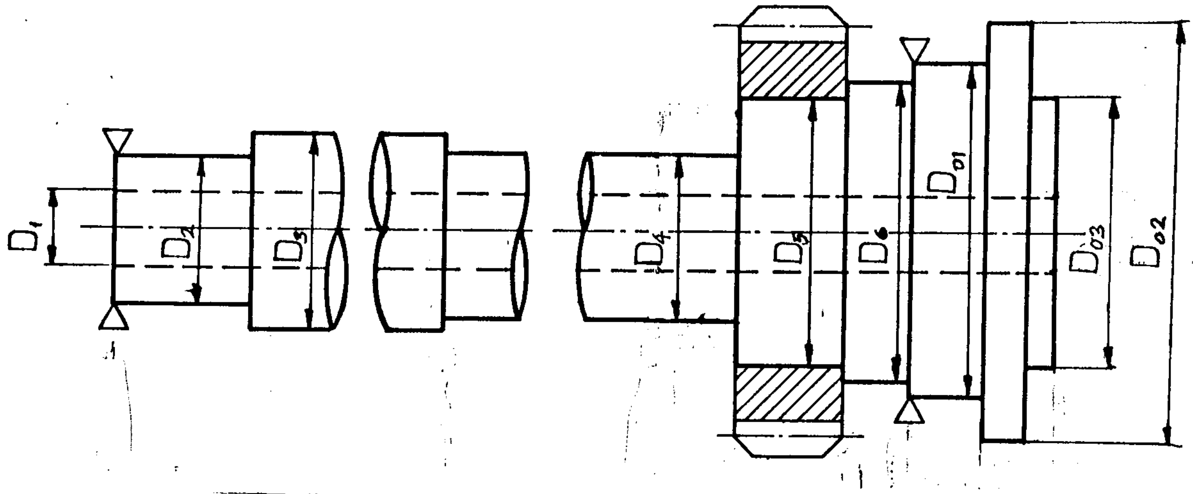


Fig.5.5. Variation of Total Radial Deflection due to the Span



TRIAL NO	OUT.DIA.	TOT.R.DEF.
7	X(1) = 80.00000(MM)	Y(1) = 0.15271(MM)
6	X(2) = 90.00000(MM)	Y(2) = 0.06316(MM)
5	X(3) =100.00000(MM)	Y(3) = 0.03180(MM)
1	X(4) =110.00000(MM)	Y(4) = 0.01720(MM)
2	X(5) =120.00000(MM)	Y(5) = 0.00938(MM)
3	X(6) =130.00000(MM)	Y(6) = 0.00480(MM)
4	X(7) =150.00000(MM)	Y(7) = 0.00008(MM)

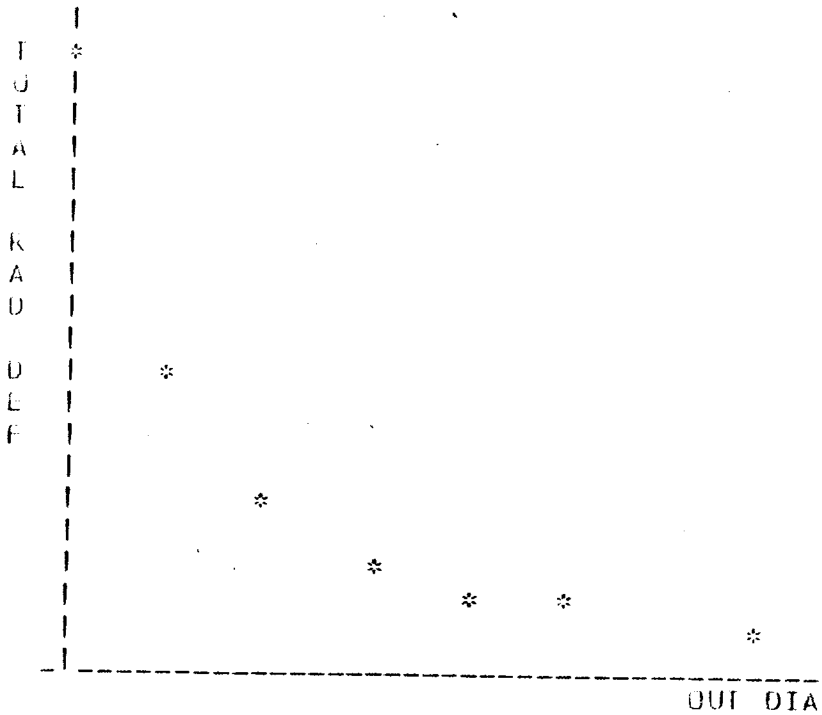
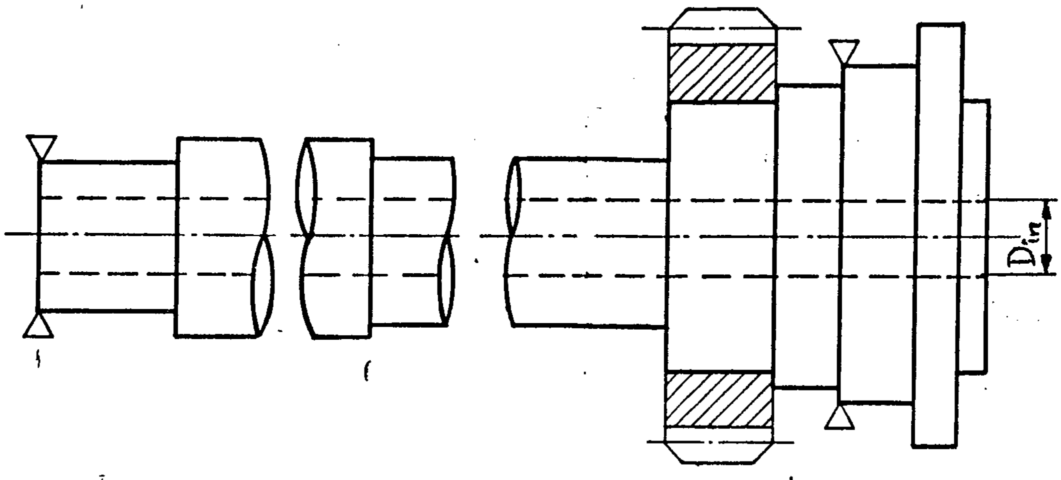


Fig.5.6. Variation of Radial Deflection at Nose due to the Outer Diameter.



TRIAL NO	IN.DIA.	TOT.R.DEF.
7	X(1) = 10.00000(MM)	Y(1) = 0.02218(MM)
6	X(2) = 10.00000(MM)	Y(2) = 0.02218(MM)
5	X(3) = 20.00000(MM)	Y(3) = 0.02225(MM)
4	X(4) = 30.00000(MM)	Y(4) = 0.02254(MM)
3	X(5) = 40.00000(MM)	Y(5) = 0.02337(MM)
2	X(6) = 50.00000(MM)	Y(6) = 0.02532(MM)
1	X(7) = 50.00000(MM)	Y(7) = 0.02532(MM)

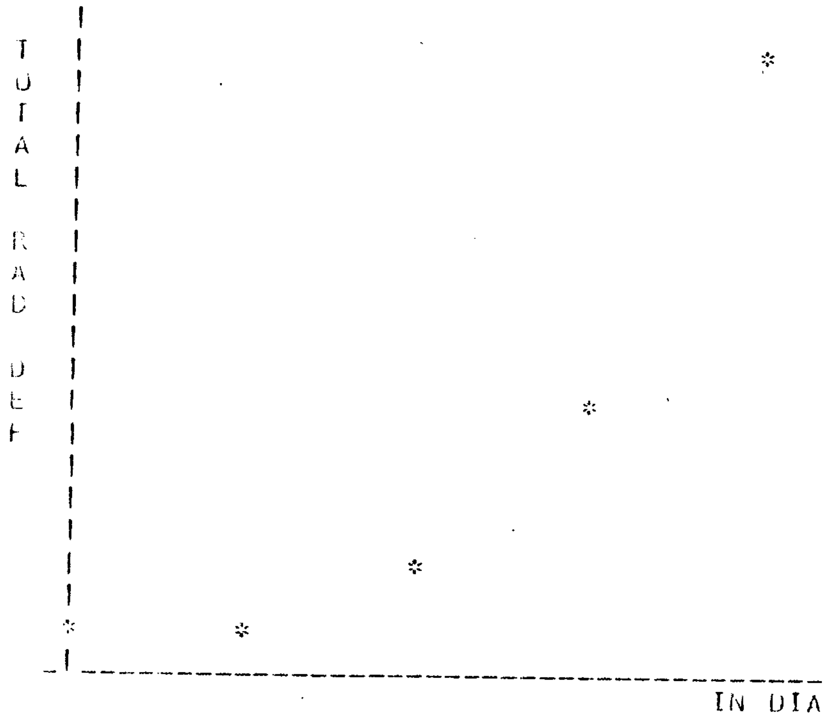
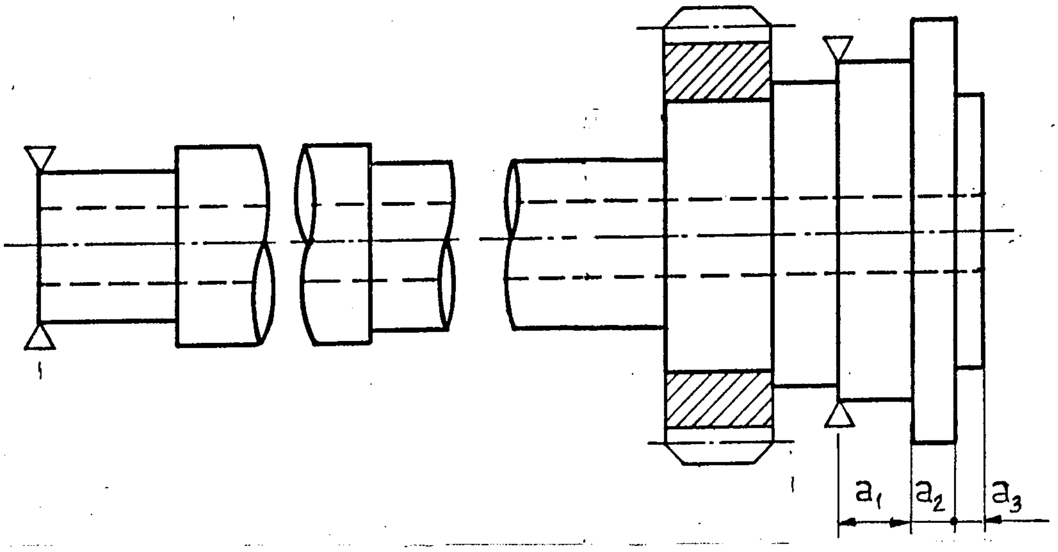


Fig.5.7. Variation of Radial Deflection at Nose due to the Inner Diameter.



TRIAL NO	OVRHNG	TOT.R.DEF.
1	X(1) = 20.00000(MM)	Y(1) = 0.00098(MM)
2	X(2) = 30.00000(MM)	Y(2) = 0.00009(MM)
3	X(3) = 45.00000(MM)	Y(3) = 0.00180(MM)
4	X(4) = 60.00000(MM)	Y(4) = 0.00438(MM)
7	X(5) =105.00000(MM)	Y(5) = 0.01663(MM)
5	X(6) =135.00000(MM)	Y(6) = 0.02900(MM)
6	X(7) =195.00000(MM)	Y(7) = 0.06535(MM)
8	X(8) =231.00000(MM)	Y(8) = 0.09556(MM)
10	X(9) =255.00000(MM)	Y(9) = 0.11959(MM)
9	X(10) =300.00000(MM)	Y(10) = 0.17381(MM)

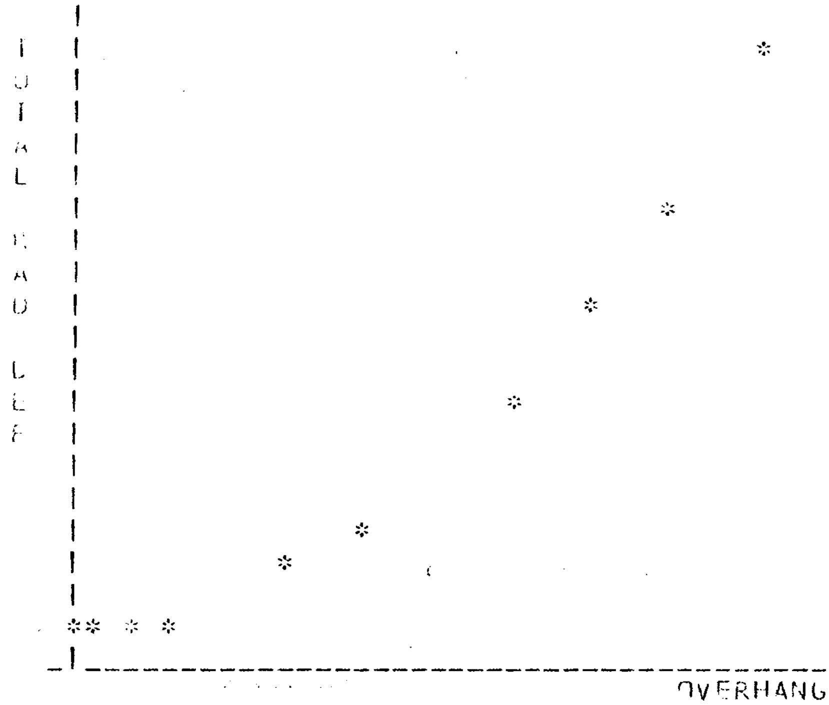
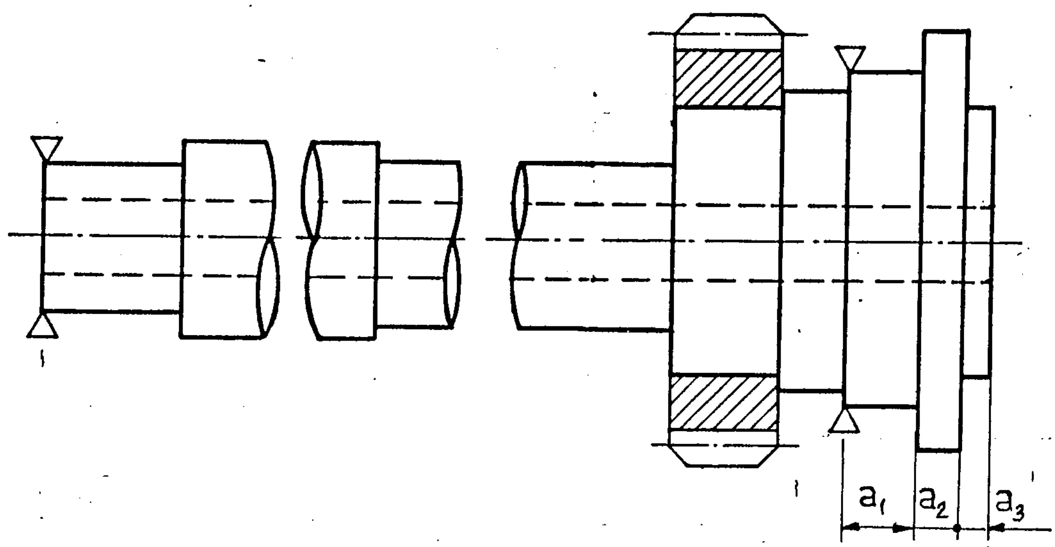


Fig.5.8. Variation of Radial Deflection at Nose due to the Overhang.



TRIAL NO	OVRHNG	TOT.AX.DEF
1	X(1) = [illegible]0.00000(MM)	Y(1) = 0.00163(MM)
2	X(2) = 30.00000(MM)	Y(2) = 0.00165(MM)
3	X(3) = 45.00000(MM)	Y(3) = 0.00168(MM)
4	X(4) = 60.00000(MM)	Y(4) = 0.00171(MM)
7	X(5) = 105.00000(MM)	Y(5) = 0.00179(MM)
8	X(6) = 13[illegible].00000(MM)	Y(6) = 0.00184(MM)
5	X(7) = 1[illegible].00000(MM)	Y(7) = 0.00195(MM)
5	X(8) = [illegible]31.00000(MM)	Y(8) = 0.00202(MM)
10	X(9) = [illegible]5[illegible].00000(MM)	Y(9) = 0.00206(MM)
9	X(10) = [illegible]00.00000(MM)	Y(10) = 0.00214(MM)

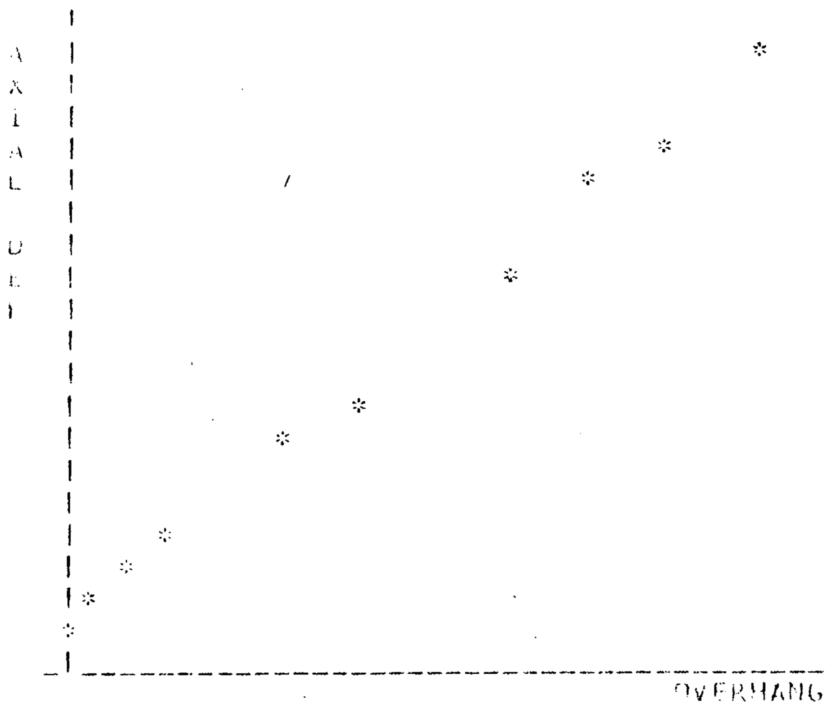
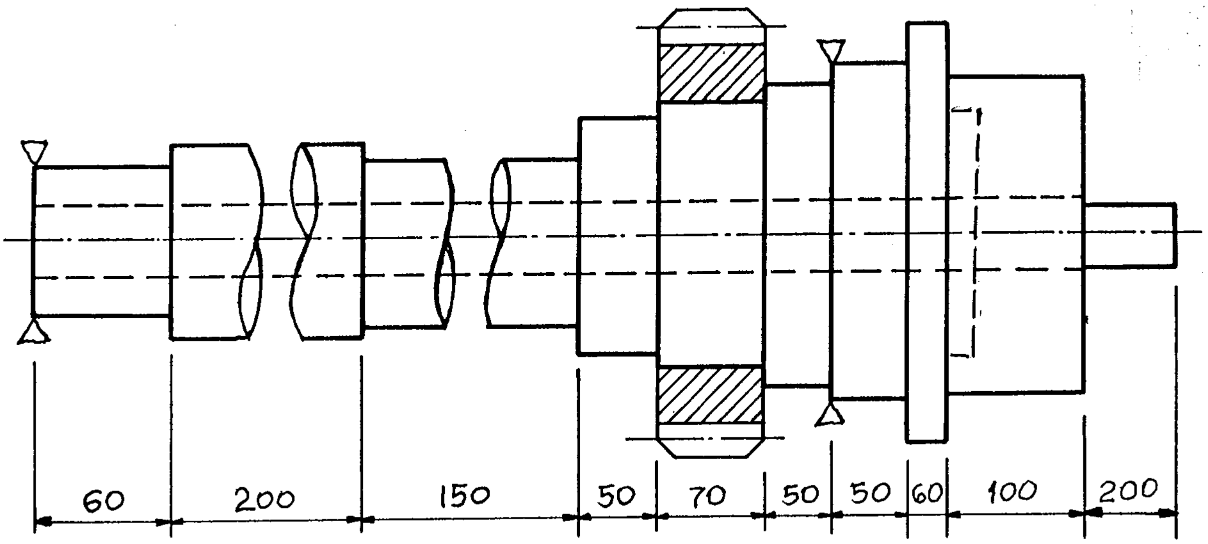


Fig.5.9. Variation of Axial Deflection due to the Overhang




```
RADIAL LOAD ON FRONT BRG.=   6038.0 (N)
AXIAL LOAD ON FRONT BRG.=   2400.0 (N)
RADIAL LOAD ON REAR BRG.=   3974.5 (N)

STRENGTH CHECK?  ANSWER(YES/NO) : NO

DEFLECTION GRADIENT ON FRONT BRG.=  0.00086(RAD)
DEFLECTION GRADIENT ON REAR BRG.= \0.00056(RAD)

RADIAL DEFL.AT NOSE DUE TO SPAN =    0.35275 (MM)
RADIAL DEFL.AT NOSE DUE TO OVERHANG =    0.23739 (MM)
RADIAL DEFL.AT NOSE DUE TO SPINDLE =    0.59015 (MM)

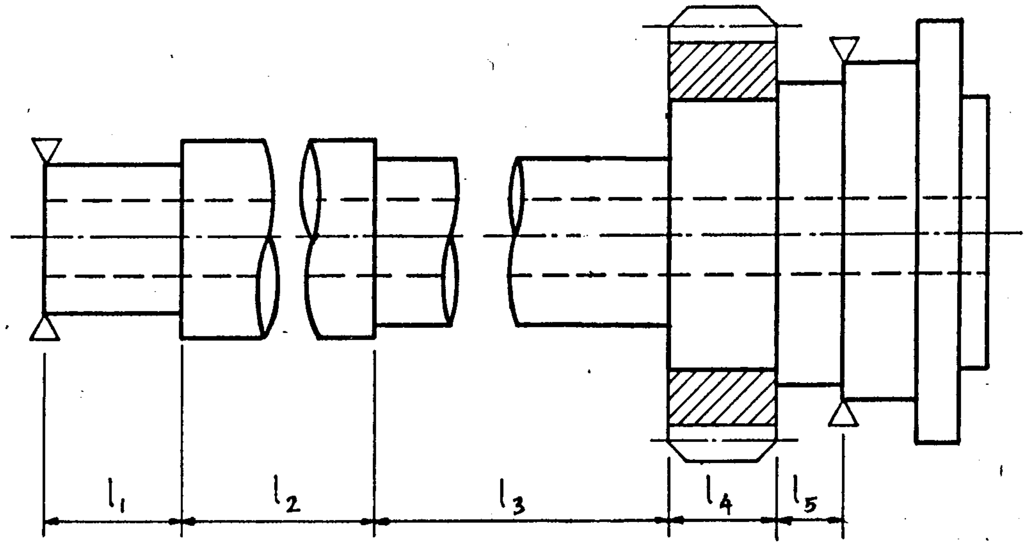
RADIAL DEFL.AT NOSE DUE TO FRONT BEARING =    0.00823 (MM)
RADIAL DEFL.AT NOSE DUE TO REAR BEARING =    0.00244 (MM)

TOTAL RADIAL DEFLECTION =    0.57948 (MM)
GROSS RADIAL STIFFNESS =          8647.95 (N/MM)

AXIAL DEFL.AT NOSE DUE TO OVERHANG =    0.00133 (MM)
AXIAL DEFL.AT NOSE DUE TO FRONT BEARING =    0.00160 (MM)

TOTAL AXIAL DEFLECTION =    0.00293 (MM)
GROSS AXIAL STIFFNESS =      819218.50 (N/MM)
```

Fig.5.10. The Spindle with Chuck and Workpiece



TRIAL NO	SPAN	TOT.DEF.
1	X(1) =100.00000(MM)	Y(1) = 0.01277(MM)
2	X(2) =150.00000(MM)	Y(2) = 0.00148(MM)
3	X(3) =200.00000(MM)	Y(3) = 0.00463(MM)
4	X(4) =250.00000(MM)	Y(4) = 0.00922(MM)
5	X(5) =300.00000(MM)	Y(5) = 0.01324(MM)
6	X(6) =350.00000(MM)	Y(6) = 0.01707(MM)
7	X(7) =400.00000(MM)	Y(7) = 0.02087(MM)
8	X(8) =450.00000(MM)	Y(8) = 0.02472(MM)
9	X(9) =500.00000(MM)	Y(9) = 0.02868(MM)
10	X(10) =600.00000(MM)	Y(10) = 0.03706(MM)

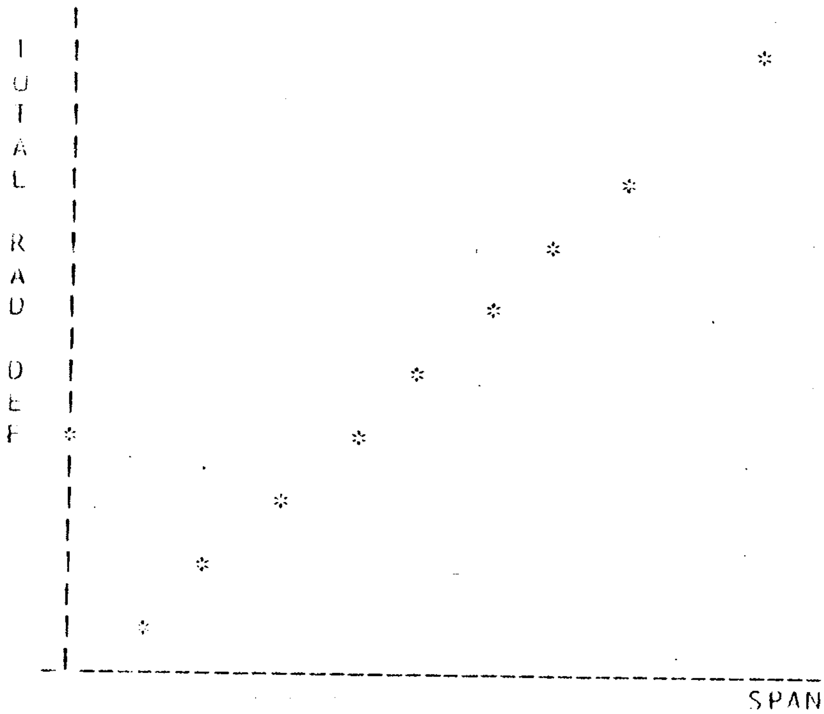
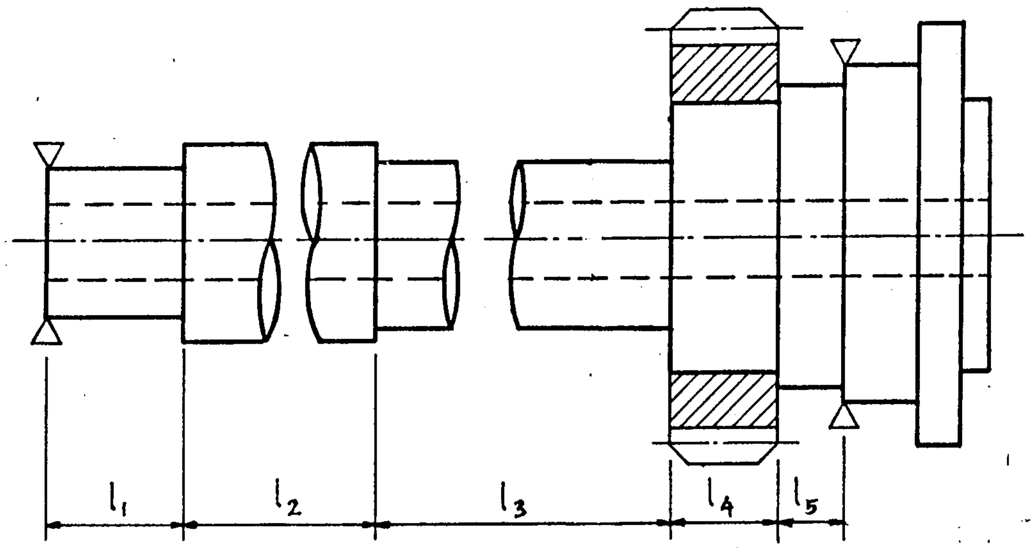


Fig.5.11. Variation of Radial Deflection at Nose due to the Span when Driving Force is on 4 th segment.



TRIAL NO	SPAN	TOT.DEF.
1	X(1) =100.00000(MM)	Y(1) = 0.01493(MM)
2	X(2) =150.00000(MM)	Y(2) = 0.00263(MM)
3	X(3) =200.00000(MM)	Y(3) = 0.00417(MM)
4	X(4) =250.00000(MM)	Y(4) = 0.00932(MM)
5	X(5) =300.00000(MM)	Y(5) = 0.01384(MM)
6	X(6) =350.00000(MM)	Y(6) = 0.01814(MM)
7	X(7) =400.00000(MM)	Y(7) = 0.02241(MM)
9	X(8) =450.00000(MM)	Y(8) = 0.02672(MM)
8	X(9) =500.00000(MM)	Y(9) = 0.03116(MM)
10	X(10) =600.00000(MM)	Y(10) = 0.04056(MM)

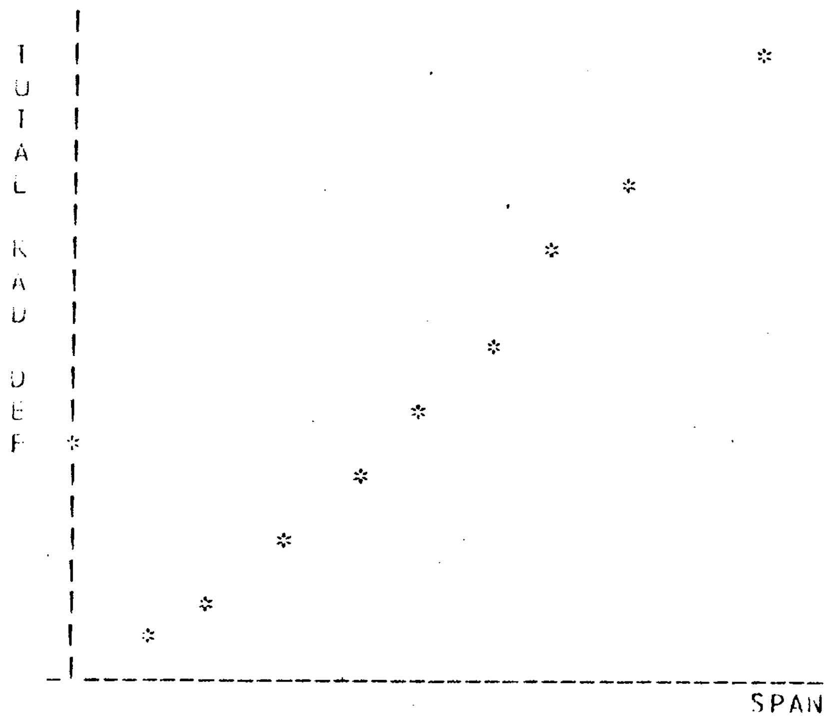
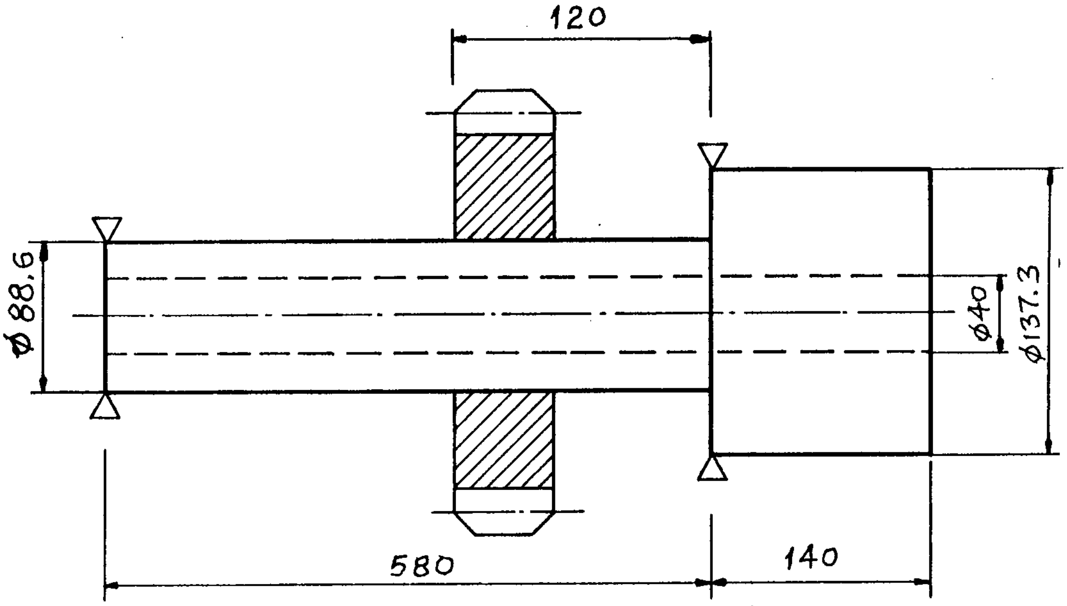


Fig.5.12. Variation of Radial Deflection at Nose due to the Span when Driving Force is on 2 nd segment




```
DEFLECTION GRADIENT ON FRONT BRG.=   0.00030(RAD)
DEFLECTION GRADIENT ON REAR BRG.=   0.00018(RAD)

RADIAL DEFLECT NOSE DUE TO SPAN =   0.04174 (MM)
RADIAL DEFLECT NOSE DUE TO OVERHANG =   0.00128 (MM)
RADIAL DEFLECT NOSE DUE TO SPINDLE =   0.04302 (MM)

RADIAL DEFLECT NOSE DUE TO FRONT BEARING =   0.00341 (MM)
RADIAL DEFLECT NOSE DUE TO REAR BEARING =   0.00033 (MM)

TOTAL RADIAL DEFLECTION =   0.03927 (MM)
GROSS RADIAL STIFFNESS =   127606.44 (N/MM)

AXIAL DEFLECT NOSE DUE TO OVERHANG =   0.00012 (MM)
AXIAL DEFLECT NOSE DUE TO FRONT BEARING =   0.00160 (MM)

TOTAL AXIAL DEFLECTION =   0.00172 (MM)
GROSS AXIAL STIFFNESS =   1375344.00 (N/MM)
```

Fig.5.13. One-step Equivalent Model

CHAPTER 6

DISCUSSION AND CONCLUSIONS

6.1. DISCUSSION

The work reported in the previous chapters has covered the modelling and design of spindles for lathes. Before drawing the final conclusions from this study, it is preferred to discuss the results presented in the previous chapter.

The length of span is an important factor which has a large influence on the nose deflection. As noticed in Figure 5.5, the span length which gives minimum deflection has no practical value if compared to the existing machines. Hence, obtaining the minimum deflection must not be the aim. The designer has to decide on the length of the span by considering the maximum deflection that can be tolerated as well as by considering its partical value.

The changes in outer and inner diameters of the spindle is obvious. Increasing outer diameter decreases the deflection at nose and vice versa. The point that has to be kept in mind that increasing outer diameter will increase the inertia of the spindle which will adversely effect the acceleration capability of the actuator.

Another factor which increases the radial deflection is the increment in the overhang length. This result implies that the overhang of the spindle should be kept as small as possible. Other design requirements specify the length of the overhang . It is seen that the existance of chuck on the spindle should be considered in spindle designs.

The position of the driving force is another important factor that must be taken into account. When the application point of driving force is moved towards the rear bearing, the nose deflection will increase. Consideration must be given in determining the position of the gear which drives the spindle.

The program performs the calculations which may be for a specific condition where all the design parameters are fixed. It also performs the calculations which may be for various conditions where one of the design parameters is varied while the rest are kept constant. Integration of the graphics program into CAD package has made easier the investigation of the effects of the parameters on the nose deflection.

One thing that must be kept in mind that with the calculations and the plots, this package aids the designer in his decision in finalising the design. The designer, using his past experience, makes use of the results obtained by this package and reaches to the best solution among several alternatives.

6.2. CONCLUSIONS

A systematic approach is developed for the design of the spindles. This study does not provide the designer with a fully automized process of optimization. It does, however, provide a means of analysis wherein he may combine the answers obtained from the computer with the human decision-making process. The result here is an economical and reliable engineering approach to the optimization of a spindle design.

A computer-aided design package has been prepared. It consists of the design of spindle which is one of the main components of a machine-tool. So, it provide a further step in the design of machine-tools. The package is user-oriented. Informative and error messages are inserted into the program to aid the designer in his judgements.

A general formula is developed to determine the radial deflection of spindles. This formula gives the designer an opportunity of investigating multi-step models as well as one step model.

A model to determine critical speed of the spindles is proposed. It is applicable to any other shaft. The package may be used for shafts in similar configuration.

6.3. SUGGESTIONS FOR FUTURE WORK

The work presented in this thesis can be extended to the following fields to provide many options to the machine tool designer.

1- After surveying the force analysis of the spindles of other machine-tools, the spindle design may be extended to all machine-tools.

2- A detailed analysis of rolling element bearings under combined loading should be carried out.

3- Computer-aided selection program for antifriction bearings should be integrated into this package.

4- CAD package for Gear Boxes should be prepared and integrated into this package to design the complete main drive unit.

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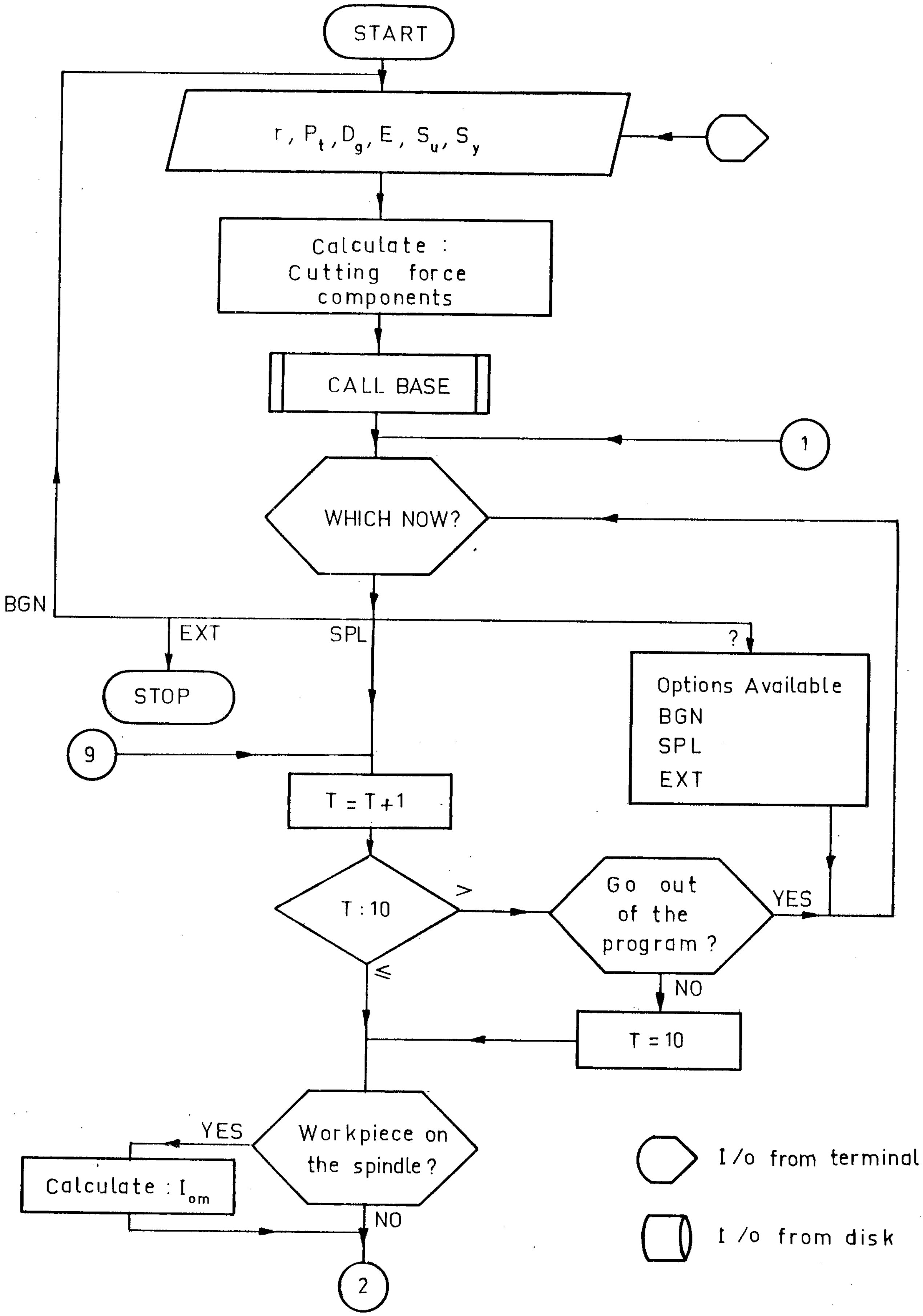
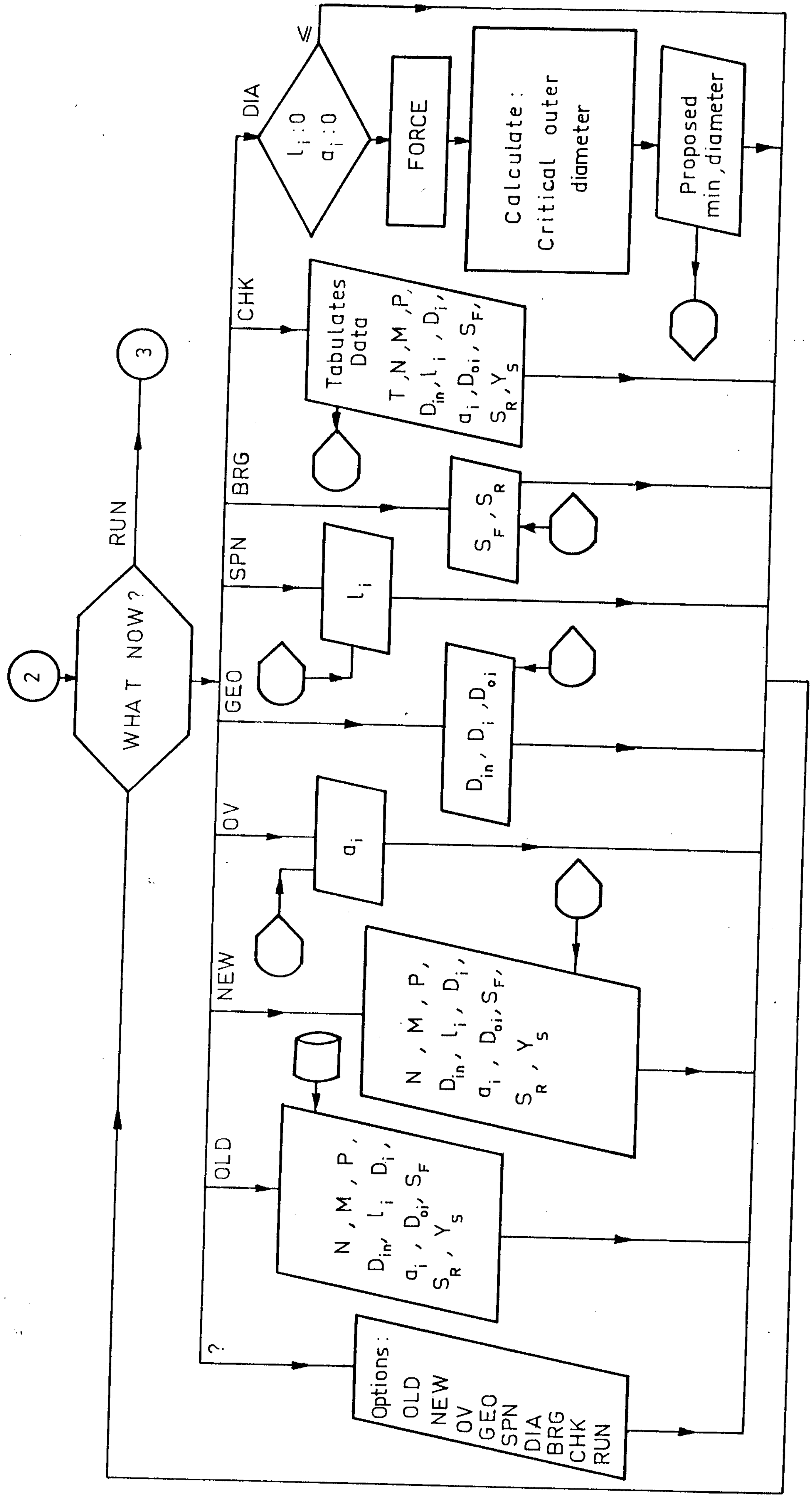
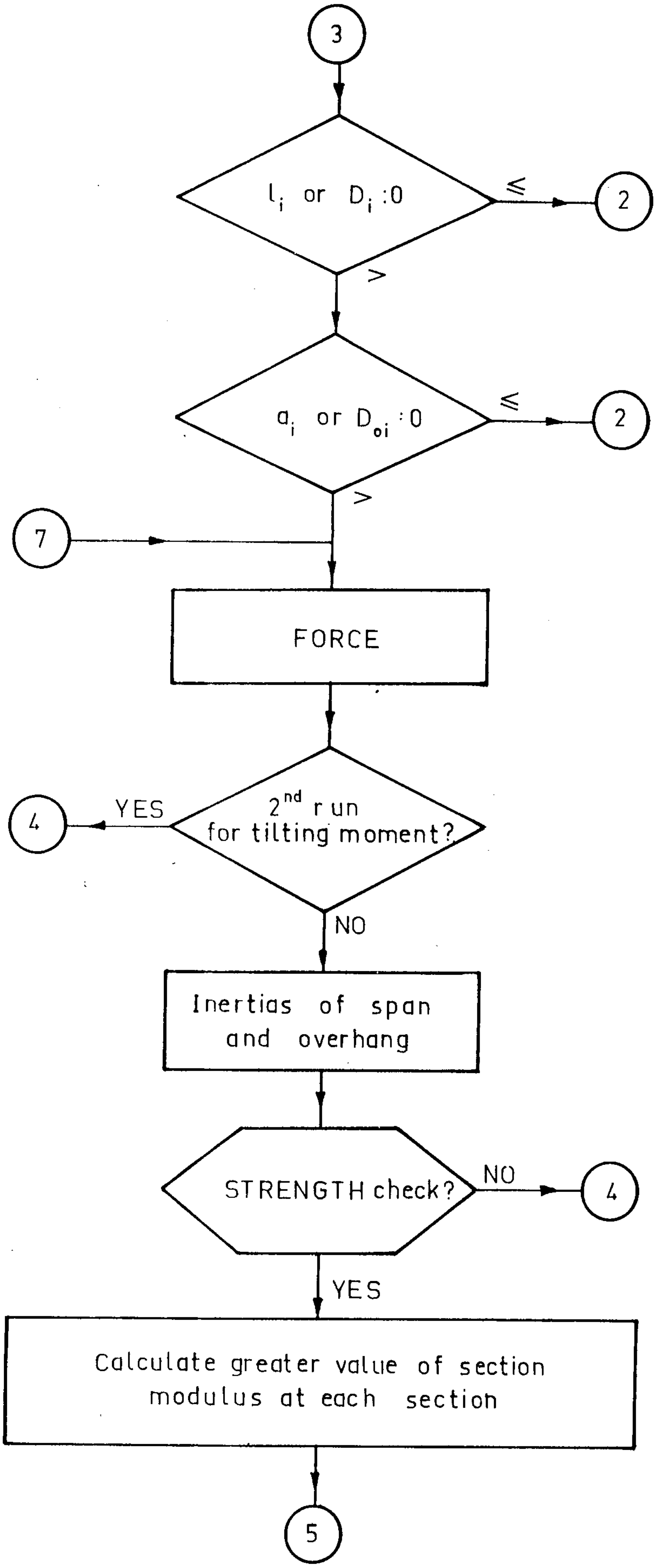
APPENDIX 5.1.

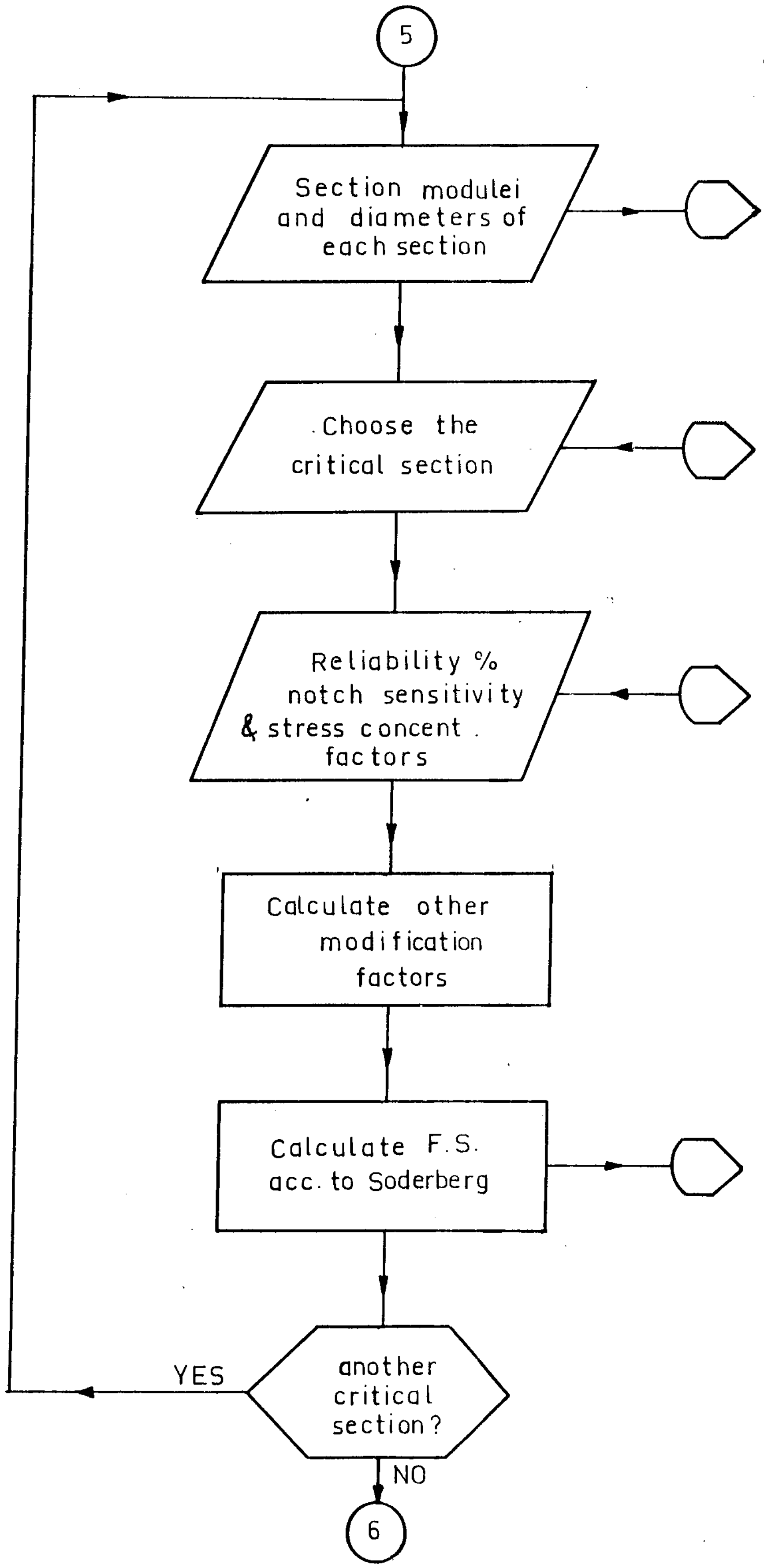
FLOWCHART OF THE PROGRAM

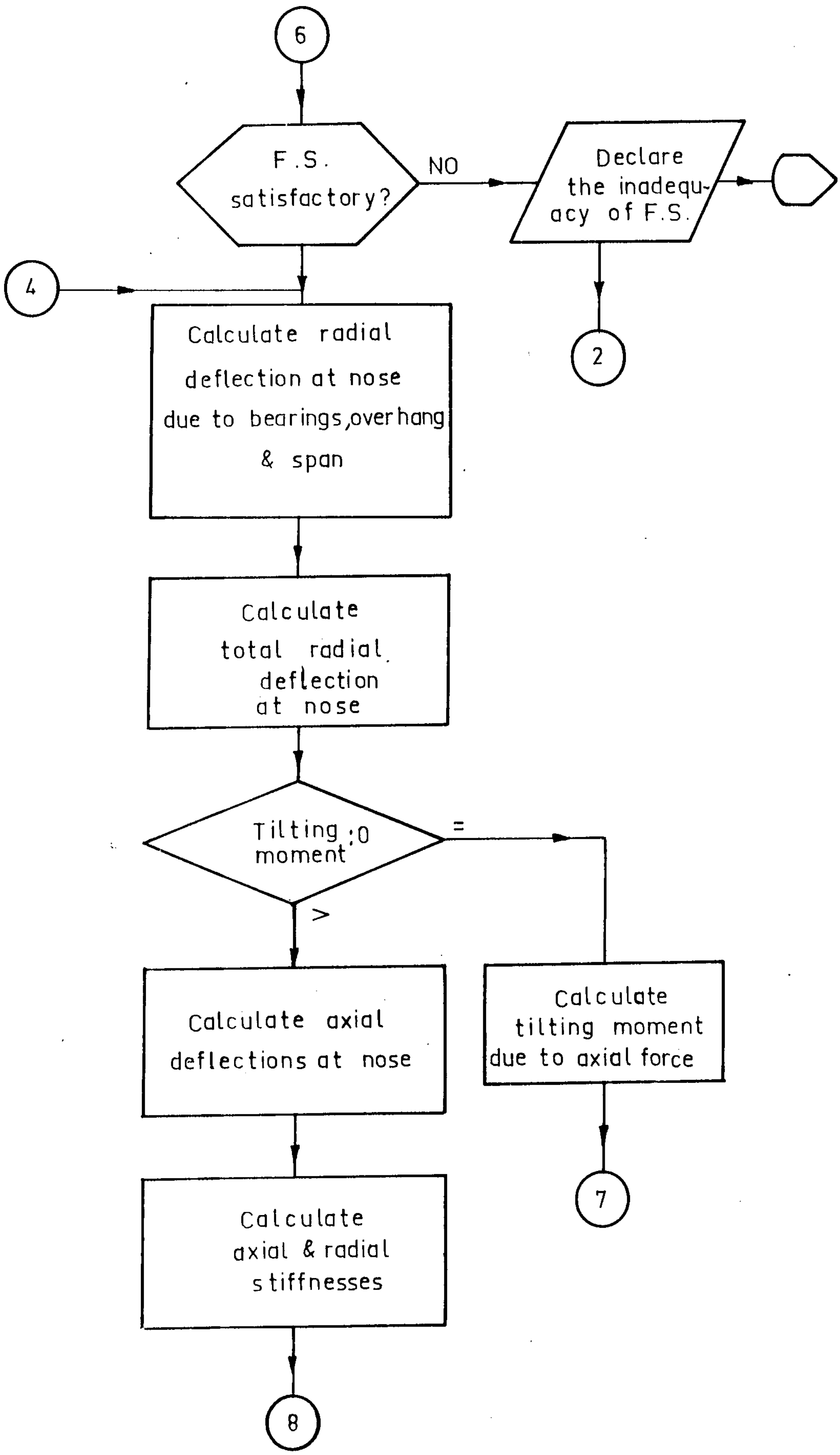


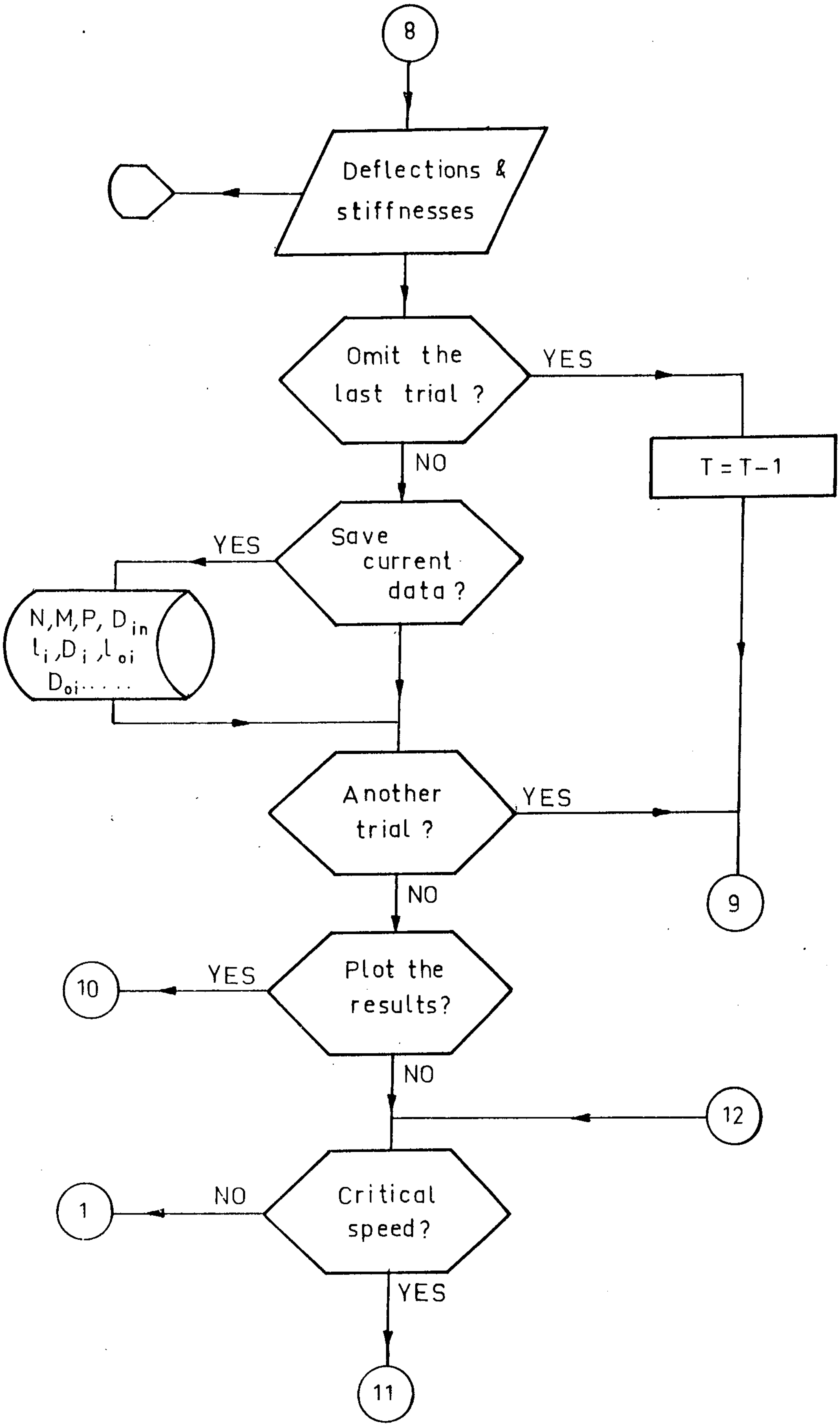


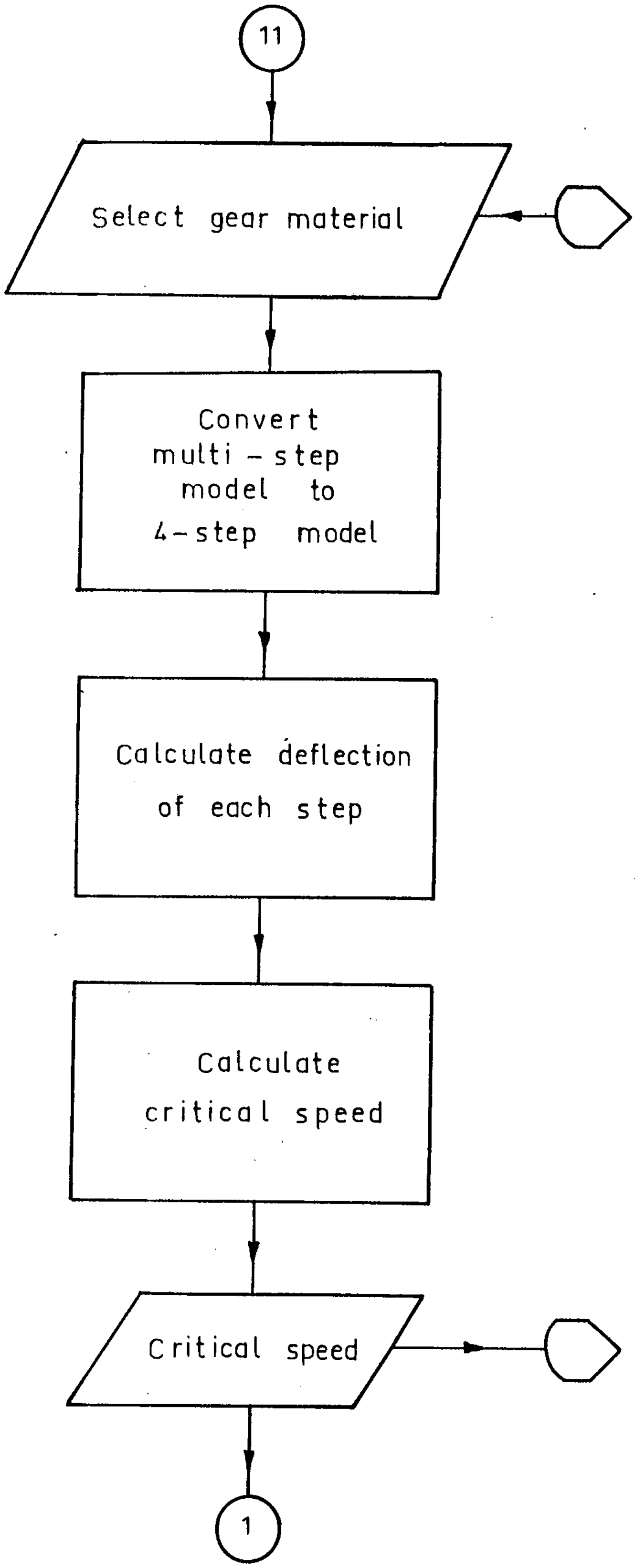
2
WHAT NOW?
RUN
3
?
OLD
NEW
OV
GEO
SPN
BRG
CHK
DIA
Options: OLD NEW OV GEO SPN DIA BRG CHK RUN
N, M, P, D_{in}, l_i, D_i, a_i, D_{oi}, S_F, S_R, Y_S
N, M, P, D_{in}, l_i, D_i, a_i, D_{oi}, S_F, S_R, Y_S
a_i
D_{in}, D_i, D_{oi}
l_i
S_F, S_R
Tabulates Data T, N, M, P, D_{in}, l_i, D_i, a_i, D_{oi}, S_F, S_R, Y_S
$l_i : 0$
$a_i : 0$
≠
FORCE
Calculate : Critical outer diameter
Proposed min, diameter

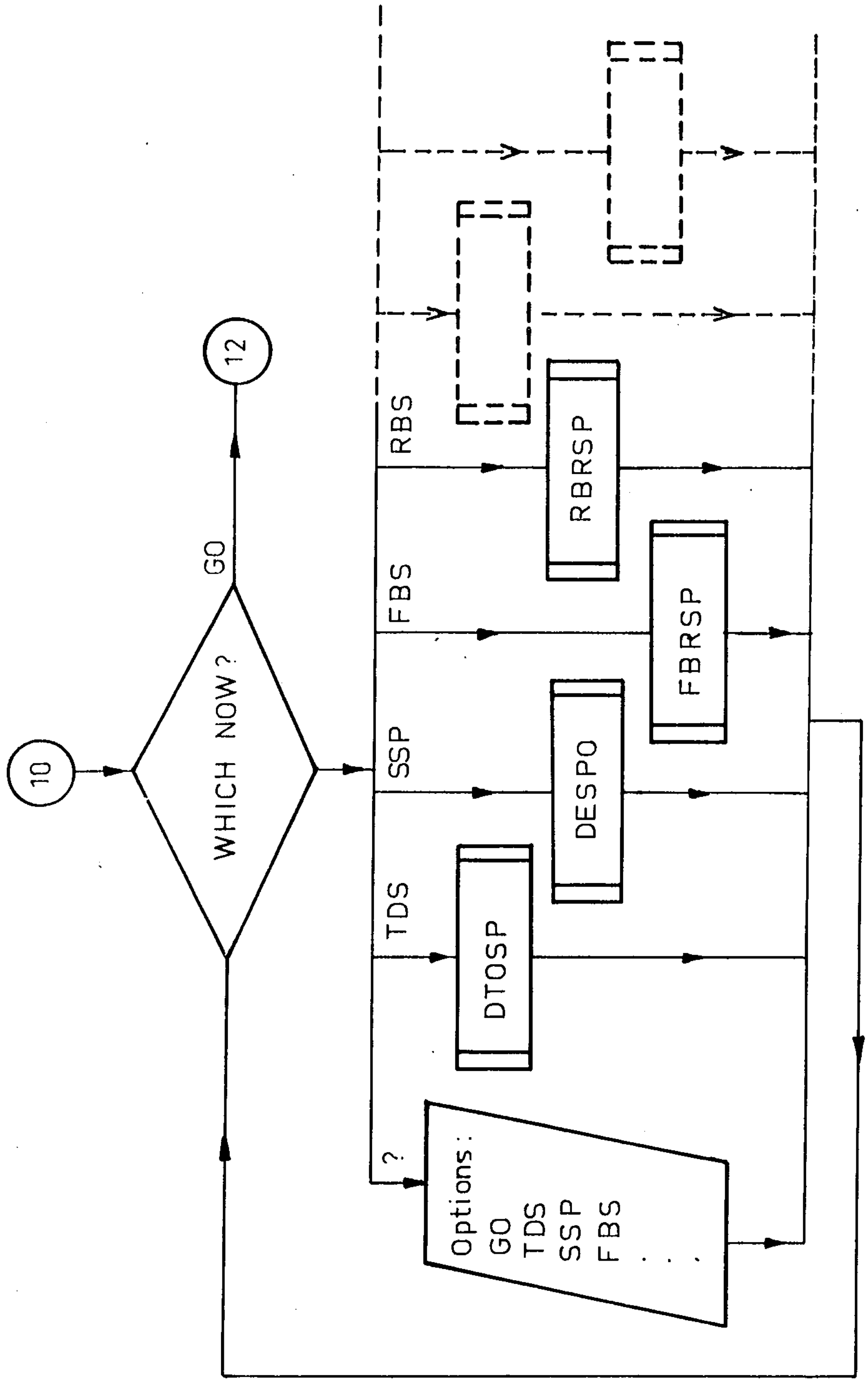

3
l_i or D_i : 0
≤
2
>
a_i or D_{oi} : 0
≤
2
>
7
FORCE
YES
4
2^{nd} run for tilting moment?
NO
Inertias of span and overhang
STRENGTH check?
NO
4
YES
Calculate greater value of section modulus at each section
5

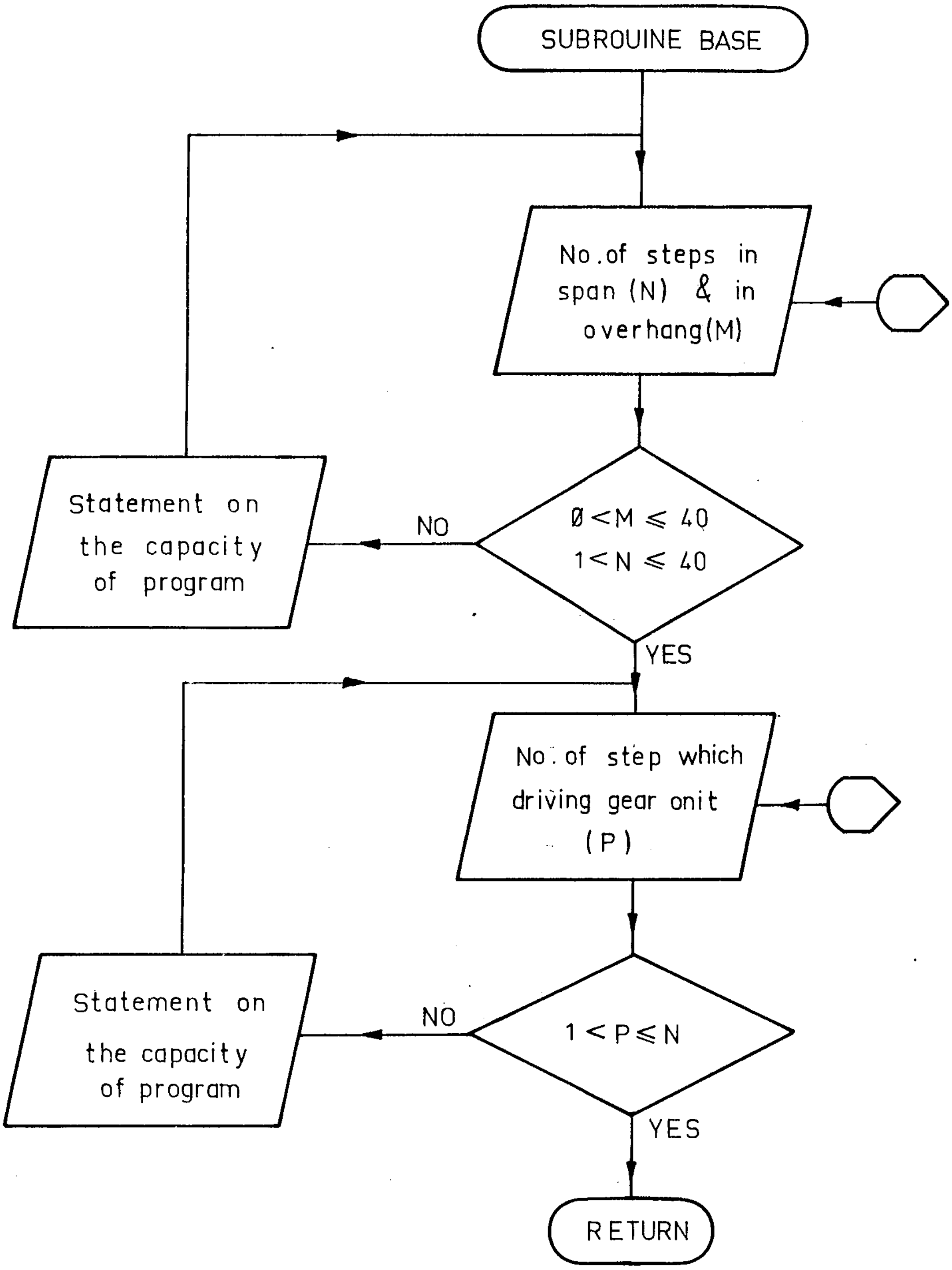

5
Section modulei and diameters of each section
Choose the critical section
Reliability %
notch sensitivity
& stress concent. factors
Calculate other modification factors
Calculate F.S. acc. to Soderberg
another critical section?
YES
NO
6

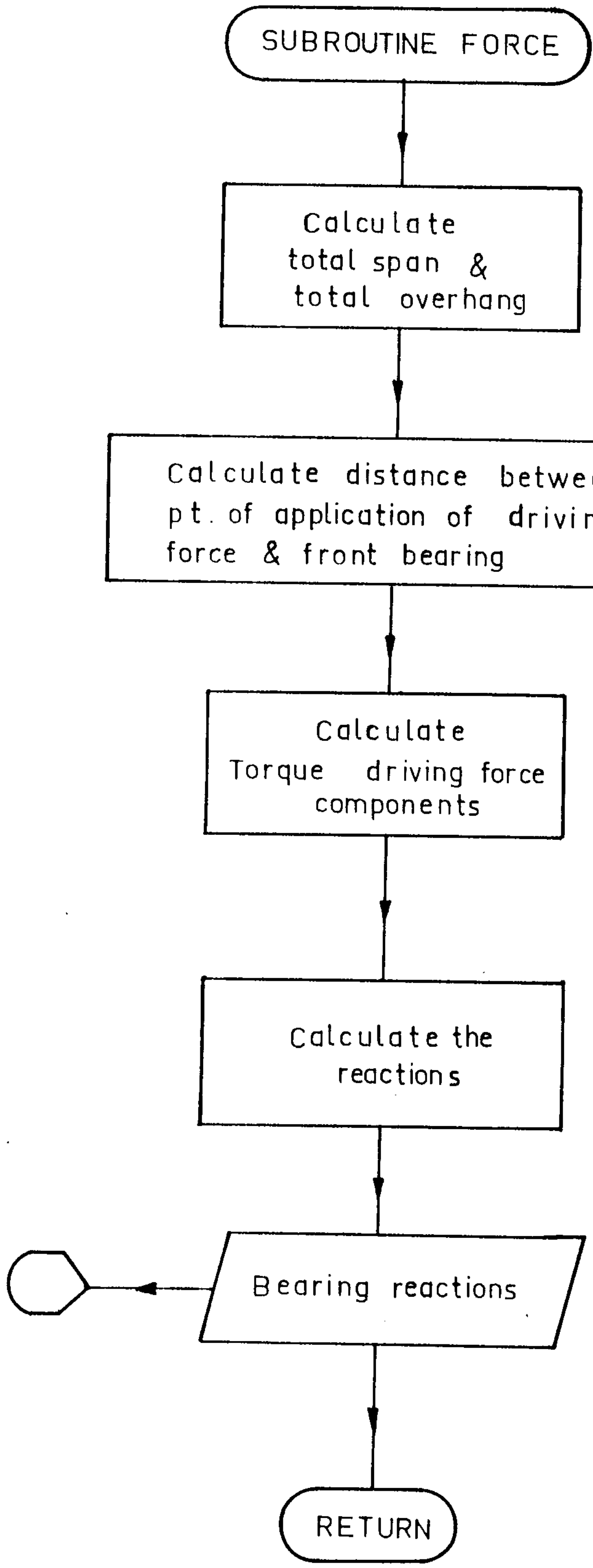

6
F.S. satisfactory?
NO
Declare the inadequacy of F.S.
2
4
Calculate radial deflection at nose due to bearings,overhang & span
Calculate total radial deflection at nose
Tilting moment :0
=
>
Calculate axial deflections at nose
Calculate tilting moment due to axial force
7
Calculate axial & radial stiffnesses
8

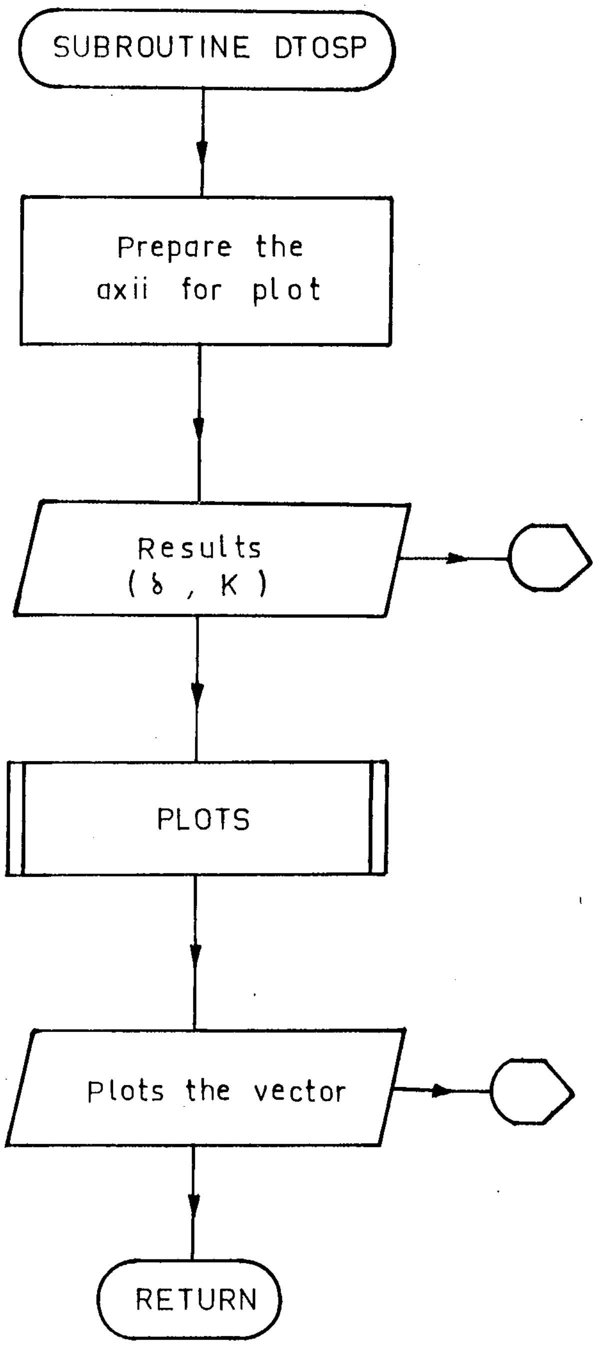

8
Deflections &
stiffnesses
Omit the
last trial ?
YES
T = T−1
NO
Save
current
data ?
YES
N,M,P, D_{in}
l_i ,D_i ,l_{oi}
D_{oi}.....
Another
trial ?
YES
NO
9
Plot the
results?
YES
10
NO
12
Critical
speed?
NO
1
YES
11

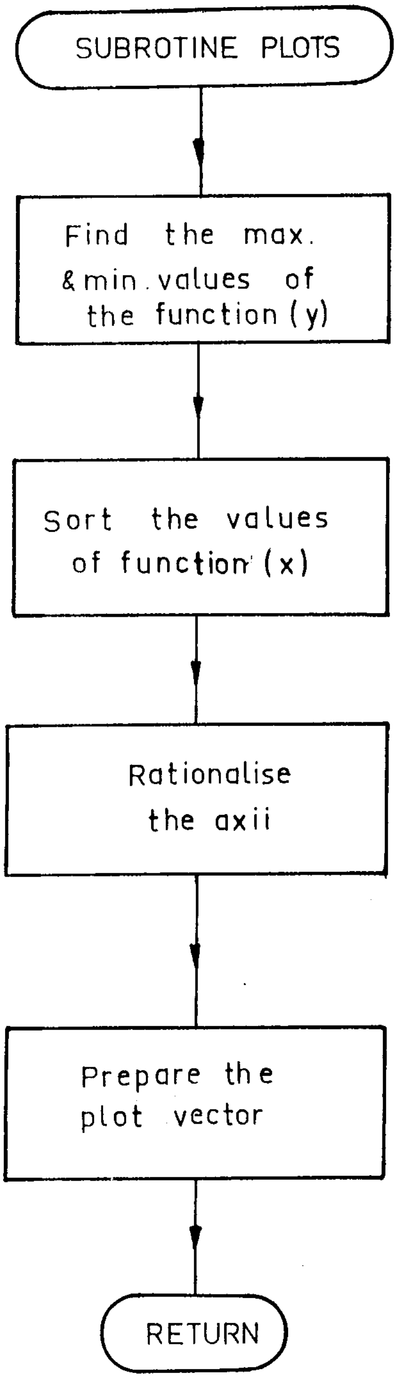

11
Select gear material
Convert multi - step model to 4-step model
Calculate deflection of each step
Calculate critical speed
Critical speed
1


10
WHICH NOW?
GO
12
TDS
SSP
FBS
RBS
?
DTOSP
DESPO
FBRSP
RBRSP
Options:
GO
TDS
SSP
FBS
. . .


SUBROUINE BASE
No.of steps in span (N) & in overhang(M)
Statement on the capacity of program
NO
Ø<M ≤ 40
1<N ≤ 40
YES
No. of step which driving gear onit (P)
Statement on the capacity of program
NO
1<P≤N
YES
RETURN


SUBROUTINE FORCE
Calculate total span & total overhang
Calculate distance between pt. of application of driving force & front bearing
Calculate Torque driving force components
Calculate the reactions
Bearing reactions
RETURN


SUBROUTINE DTOSP
Prepare the axii for plot
Results (ɣ , K)
PLOTS
Plots the vector
RETURN


SUBROTINE PLOTS
Find the max. & min. values of the function (y)
Sort the values of function (x)
Rationalise the axii
Prepare the plot vector
RETURN

APPENDIX 5.2

ILLUSTRATION OF FLOW OF THE PROGRAM

```
GIVE RADIUS OF THE WORKPIECE (MM)
GIVE TANGENTIAL COMPONENT OF CUTTING FORCE (N)
PITCH CIRCLE DIAMETER OF THE GEAR ON THE SPINDLE (MM)
GIVE MODULUS OF ELASTICITY OF SPINDLE (N/SQMM)
GIVE ULTIMATE STRENGTH OF SPINDLE MATERIAL (N/SQMM)
GIVE YIELD STRENGTH OF SPINDLE MATERIAL (N/SQMM)


RADIUS OF THE WORKPIECE =   50.00     (MM)
TANGENTIAL COMP. OF CUTTING FORCE =   4800.0    (N)
PITCH DIAMETER OF GEAR ON SPINDLE = 170.00    (MM)
MOD.OF ELASTICITY OF THE MATERIAL = 0.2070E+06    (N/SQMM)
ULTIMATE STRENGTH OF THE MATERIAL = 0.1700E+04    (N/SQMM)
YIELD STRENGTH OF THE MATERIAL = 0.1400E+04    (N/SQMM)

-------------------------------------------------------------

TRIAL NO.:  1


GIVE NO.OF STEPS IN SPAN AND
NO.OF STEPS IN OVERHANG, RESPECTIVELY

NO.OF STEPS IN SPAN   =    6
NO.OF STEPS IN OVERHANG =   3

GIVE NO.OF STEP, WHICH DRIVING GEAR ON IT

NO.OF STEP, WHICH DRIVING GEAR ON IT =   5

TYPE  ?  FOR HELP

WHICH NOW ? : NEW


OPTIONS AVAILABLE :

  BGN   CHANGES PREVIOUS DATA
  SPL   ANALYSES OF SPINDLE
  EXT   EXITS THE PROGRAM


TYPE  ?  FOR HELP

WHICH NOW ? : SPL
```

```
TYPE  ?  FOR HELP

WHAT NOW ? : ?


OPTIONS AVAILABLE :

   OLD   INPUTS PREVIOUSLY RECORDED DATA
   NEW   INPUTS NEW DATA
   GEO   CHANGES DIAMETERS OF SPINDLE
   SPN   CHANGES SPAN LENGTHS
   OV    CHANGES OVERHANG LENGTHS
   DIA   FOR INITIAL GUESS OF DIAMETER
   BRG   CHANGES BEARING(S)
   CHK   TABULATES DATA FOR CHECK
   RUN   RUNS THE PROGRAM


WHAT NOW ? : NEW


GIVE  LENGTHS OF SPAN PIECES IN (MM)
60.0000000
200.000000
150.000000
50.0000000
70.0000000
50.0000000

GIVE  LENGTHS OF OVERHANG PIECES IN (MM)
50.0000000
60.0000000
20.0000000

GIVE  INNER DIAMETER OF THE SPINDLE IN (MM)
40.0000000

GIVE  OUTER DIAMETERS OF SPAN PIECES IN (MM)
80.0000000
85.0000000
84.0000000
88.0000000
92.0000000
110.000000

GIVE  OUTER DIAMETERS OF OVERHANG PIECES IN (MM)
120.000000
160.000000
110.000000
```

```
WHAT NOW ? : BRG

GIVE RADIAL STIFFNESSES OF
FRONT & REAR BEARINGS IN (N/MM)
1500000.00          1500000.00
GIVE AXIAL STIFFNESS OF FRONT BEARING IN (N/MM)
1500000.00

WHAT NOW ? : CHK


     TRIAL NO. =  1
  NO.OF STEPS IN SPAN =  6
  NO.OF STEPS IN OVERHANG =  3
  NO.OF STEP, GEAR ON IT =  5

  INNER DIAMETER =  40.000  (MM)

LENGTH OF SPAN PIECE ( 1) =   60.000  (MM)
LENGTH OF SPAN PIECE ( 2) = 200.000  (MM)
LENGTH OF SPAN PIECE ( 3) = 150.000  (MM)
LENGTH OF SPAN PIECE ( 4) =   50.000  (MM)
LENGTH OF SPAN PIECE ( 5) =   70.000  (MM)
LENGTH OF SPAN PIECE ( 6) =   50.000  (MM)


OUTER DIA.OF SPAN PIECE ( 1) =  90.000  (MM)
OUTER DIA.OF SPAN PIECE ( 2) =  85.000  (MM)
OUTER DIA.OF SPAN PIECE ( 3) =  84.000  (MM)
OUTER DIA.OF SPAN PIECE ( 4) =  88.000  (MM)
OUTER DIA.OF SPAN PIECE ( 5) =  92.000  (MM)
OUTER DIA.OF SPAN PIECE ( 6) = 110.000  (MM)


LENGTH OF OVERHANG PIECE ( 1) =  50.000  (MM)
LENGTH OF OVERHANG PIECE ( 2) =  60.000  (MM)
LENGTH OF OVERHANG PIECE ( 3) =  20.000  (MM)


OUTER DIA.OF OVERHANG PIECE ( 1) = 120.000 (MM)
OUTER DIA.OF OVERHANG PIECE ( 2) = 160.000 (MM)
OUTER DIA.OF OVERHANG PIECE ( 3) = 110.000 (MM)

FRONT BRG.RADIAL STIFFNESS =  0.1500000E+07 (N/MM)

REAR BRG.RADIAL STIFFNESS =  0.1500000E+07 (N/MM)

FRONT BRG.AXIAL STIFFNESS =  0.1500000E+07 (N/MM)
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WHAT NOW ? : RUN

RADIAL LOAD ON FRONT BRG.=  3618.7 (N)
AXIAL LOAD ON FRONT BRG.=  2400.0 (N)
RADIAL LOAD ON REAR BRG.=  1555.2 (N)

STRENGTH CHECK?  ANSWER(YES/NO) : YES

CHOOSE THE CRITICAL SECTION


SPAN :
SECTION 1 :  M/I=  0.0495048203   T/J=  0.0000000000   DIAMETER= 80.0
SECTION 2 :  M/I=  0.1744253040   T/J=  0.0000000000   DIAMETER= 85.0
SECTION 3 :  M/I=  0.2750552890   T/J=  0.0000000000   DIAMETER= 84.0
SECTION 4 :  M/I=  0.2538642290   T/J=  0.0000000000   DIAMETER= 88.0
SECTION 5 :  M/I=  0.1966040730   T/J=  0.0353886330   DIAMETER= 92.0
SECTION 6 :  M/I=  0.0944127440   T/J=  0.0169942454   DIAMETER=110.0

OVERHANG :
SECTION 1 :  M/I=  0.0398788527   T/J=  0.0119366273   DIAMETER=120.0
SECTION 2 :  M/I=  0.0141939297   T/J=  0.0169942454   DIAMETER=160.0

TYPE THE M/I VALUE ; T/J VALUE AND
DIAMETER OF THE CRITICAL SECTION, RESPECTIVELY
FOR POSSIBLE CRITICAL SECTION  :
M/I=  0.2750549910     T/J=  0.0000000000     DIAMETER= 84.000


REQUIRED RELIABILITY =0.999899983

NOTCH SENSITIVITY(Q)=0.950

STRESS CONCENTRATION FACTOR(KT)=1.950


FACTOR OF SAFETY(ACC.TO SODERBERG) =          14.9243
FACTOR OF SAFETY(ACC.TO MISES_HENCKY) =          14.9243


DO YOU WANT TO TRY
ANOTHER CRITICAL SECTION? ANSWER(YES/NO) : NO

IS FACTOR OF SAFETY SATISFACTORY ? : YES

DEFLECTION GRADIENT ON FRONT BRG.=  0.00026(RAD)
DEFLECTION GRADIENT ON REAR BRG.= \0.00017(RAD)

RADIAL DEFL.AT NOSE DUE TO SPAN =   0.03361 (MM)
RADIAL DEFL.AT NOSE DUE TO OVERHANG =   0.00149 (MM)
RADIAL DEFL.AT NOSE DUE TO SPINDLE =   0.03509 (MM)

RADIAL DEFL.AT NOSE DUE TO FRONT BEARING =   0.00301 (MM)
RADIAL DEFL.AT NOSE DUE TO REAR BEARING =   0.00024 (MM)

TOTAL RADIAL DEFLECTION =    0.03184 (MM)
GROSS RADIAL STIFFNESS =      157398.25 (N/MM)
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AXIAL DEFL.AT NOSE DUE TO OVERHANG =    0.00012 (MM)
AXIAL DEFL.AT NOSE DUE TO FRONT BEARING =    0.00160 (MM)

TOTAL AXIAL DEFLECTION =    0.00172 (MM)
GROSS AXIAL STIFFNESS =     1393171.00 (N/MM)


OMIT THE LAST TRIAL ? :    NO

SAVE CURRENT DATA ? :    YES

ANOTHER TRIAL ? (YES/NO) : NO

PLOT THE RESULTS ?  ANSWER(YES/NO) : NO

DO YOU WANT TO CALCULATE CRITICAL SPEED?
ANSWER YES/NO :    YES

IF GEAR MATERIAL IS :
   BRASS :          TYPE BR
   CAST IRON :      TYPE CI
   STEEL :          TYPE ST
   ANY OTHER :      TYPE OT

GEAR MATERIAL IS : ST

CRITICAL SPEED OF THE SPINDLE IS :  11948.1 (RPM)

TYPE  ?  FOR HELP

WHICH NOW ? : ?
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