COMPUTER AIDED DESIGN OF GEARBOX KINEMATICAL ARRANGEMENT DIAGRAMS

A MASTER'S THESIS

in

MECHANICAL ENGINEERING

MIDDLE EAST TECHNICAL UNIVERSITY

By

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September, 1985

44088

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ABSTRACT

COMPUTER-AIDED DESIGN OF GEARBOX KINEMATICAL ARRANGEMENT DIAGRAMS

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M.S. in Mechanical Engineering

Supervisor: Asst. Prof. Dr. I. Huseyin Filiz

September, 1985, 120 pages

In this thesis analytical method of obtaining kinematical arrangement diagrams is reviewed and a computer aided design package is prepared to obtain all possible kinematical arrangements of gearboxes within specified limitations on percentage error of gear ratios, on minimum and maximum transmission ratios, on gear teeth numbers and on percentage error of output speeds.

By providing the necessary input data to the program number of transmissions in each group, power of progression ratio of each transmission, output speeds and percentage errors on output speeds are calculated for each possible kinematical arrangements. In addition to these proportional elasticity and inertia of gear trains for each output speed are calculated to be used as a selection criteria between the options.

The use of program is illustrated and the results obtained are compared with the examples presented in the previous literature and with the examples chosen from existing machine tools.

Key words: Kinematical arrangement diagram, gearbox, computer aided design

ÖZET

DİŞLİ KUTUSU KURULUŞ DİYAGRAMLARININ BİLGİSAYAR DESTEKLİ TASARIMI

KARSLI, Sedat

Yüksek Lisans Tezi, Makin. Müh. Bölümü

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Bu tezde kinematik kuruluş diyagramları incelenerek, belirlenen dişli oranı, minimum ve maksimum aktarım oranları, diş sayıları ve çıktı hızları üzerindeki yüzde hata limitleri içerisinde mümkün olan bütün kinematik kuruluş diyagramlarının elde edilmesi için bilgisayar destekli tasarım paketi hazırlanmıştır.

Gerekli giriş bilgilerinin programa verilmesi ile, her guruptaki aktarım sayısı, her aktarım oranının geometrik kademe katsayısı üssü, çıktı hızları ve çıktı hızlarındaki yüzde hatalar elde edilen bütün kinematik kuruluş diyagramları için hesaplanmıştır. İlave olarak tercihler arası seçim yapabilmek için orantılı esneklik ve atalet momenti diş kutusundaki hızları sağlayan her dişli tertibatı için hesaplanmıştır.

Programın kullanılması gösterilmiş ve elde edilen neticeler başka çalışmalarda verilen örneklerle ve çalışan takım tezgahları üzerindeki örneklerle karşılaştırılmıştır.

Anahtar kelimeler: kinematik kuruluş diyagramı, dişli kutusu, bilgisayar destekli tasarım.

ACKNOWLEDGEMENTS

The author is grateful to Asst. Prof. Dr. I. Huseyin Filiz for his encouragements and suggestions throughout the study of this thesis as the supervisor.

He is indebted to his parents and wife for their support and interest.

He also wishes to thank ,

Mrs. Zenaida Williams for her accurate typing,

Mr. Derek A Williams for his encouragements,

Ms. Aysegul Dalbudak for her kindly helps in technical drawings.

Ms. Ruhat Dogantan for her helps in typing the formulae

Mr. Ahmet Demircan for his advice in using the Computer facilities

TABLE OF CONTENTS

Page

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

D	Outside diameter of gear
d	Workpiece or cutting tool diameter
d max	Maximum workpiece or cutting tool diameter
d min	Minimum workpiece or cutting tool diameter
e	Elasticity of a mechanical component
ejs	Reflected elasticity of a component
I	Inertia of a component (polar moment of inertia)
Ir	Reflected inertia of a component
ijm	Ratio of shaft speed to motor speed
$i_1, i_2, i_3, \ldots i_p$	Transmission ratios in a group
imax	Maximum transmission ratio
imax lim	Maximum limiting transmission ratio
iamax, ibmax, icmax . . . irmax	Maximum transmission ratio of each group
imin	Minimum transmission ratio
imin lim	Minimum limiting transmission ratio
iamin, ibmin, icmin. . . irmin	Minimum transmission ratio of each group

K	Torsional stiffness of a component
Kjs	Reflected torsional stiffness of a component
M	Module of gear
N	Gear teeth number
n	Turning speed
ni	Input motor speed
nj	Any spindle speed
nmax	Maximum turning speed
nmin	Minimum turning speed
pa, pb, pc . . . pr	Number of transmissions in each group
a, b, c . . . . r	Group numbers
Rd	Range of diameters
Rlim	Limiting speed range ratio
Rn	Speed range ratio
Rv	Range of cutting speeds
T	Torque
ϑ	Cutting speed
ϑ_{max}	Maximum cutting speed
ϑ_{min}	Minimum cutting speed
X	Characteristic of a transmission group
Z	Total number of stepped speeds
φ	Progression ratio
η	Efficiency
θ	Angular deflection

CHAPTER I

INTRODUCTION

Gearboxes which are used as power transmission units in the main drive of machine tools are required to provide a series of stepped speeds to obtain different cutting speeds for general purpose machine tools.

Although, gearboxes can be built with different kinds of gears and shaft arrangements parallel shaft arrangement with spur or helical gears are generally used on machine tools.

Cutting speed is equal to turning speed multiplied by periphery of the workpiece or the cutting tool. Hence, accuracy of turning speed affects the cutting speed. If turning speed is out of specified limits, actual cutting speed will be lower or higher than expected value which decreases the efficiency, increases machining time or tool wear. So stepped speeds must be provided within the specified limits for optimum machining which requires optimum design of gearbox.

Design of a gearbox starts with kinematic arrangement diagram, Kinematic arrangement diagram specifies number of groups, number of transmissions in each group and transmission ratios for a given number of output speed. Integer gear teeth numbers must be found for the transmission ratios to start strength calculations, i.e.; bending strength and surface fatigue strength. With the completed strength design size of the gears are determined. Then, shaft design, bearing selection, lubricant selection and case design must be carried out. To finish with the design, torsional analysis of gear box must be carried out to deter mine response and resonant frequencies of the gearbox..

In the design procedure the first step, determination of kinematic arrangement diagram is the most critical step. Because parameters specified by kinematic arrangement diagram are all used for further design stages. Gear teeth numbers are determined according to transmission ratios. Teeth numbers influence the strength calculations, number of transmissions in the groups, order of arrangement, of groups and values of transmission ratios determines the dynamic characteristics of gearbox .

The other design stages, bending strength, and surface fatigue strength calculations and torsional analysis can be done by known methods. These design procedures are applied to the selected gear box arrangement. A proper solution, sometimes can not be found for these design stages, if a suitable kinematic arrangement diagram is not selected at the initial stage of the design process.

Traditionally, kinematic arrangement diagrams are prepared graphically and teeth numbers are found from gear ratio tables. Graphical method and use of tables take a long time to complete one kinematic arrangement diagram and it takes very long time to analyse all of the possible solutions with this method, especially for large number of shafts. For example, for a gearbox with 4 shafts, there are a total of $(3!)^2 = 36$ possible arrangements. It is 14,400 for a gearbox with 6 shafts.

Computers are used nowadays, because of their many advantages. As it is known, computers have a high speed and accuracy in mathematical calculations. The speed of computer combined with human decision is a great tool in design work. For this purpose, highly user oriented computer programs are prepared and used.

This thesis attempts to automate the determination of all possible kinematical arrangement diagrams and gear teeth numbers for gearboxes, which are used in the main drive of machine tools. For this purpose, a computer program is prepared. This program makes available all the possible kinematical arrangement diagrams of a gear box for specified limitations on transmission ratios. Gear teeth numbers are found within the specified errors. Output speeds of gearbox and percentage error on output speed are calculated on the basis of found gear teeth numbers.

General theory of kinematic arrangement diagrams and their analysis are discussed in Chapter 3. Gear teeth number calculations are also included in this chapter. The structure of computer program is explained in Chapter 4 and its use is illustrated with specific examples taken from existing machine tools.

CHAPTER 2

LITERATURE SURVEY

2.1 INTRODUCTION

The literature most relevant to this thesis are presented in this chapter.

Section 2.2 covers the literature dealing with kinematical arrangement. Section 2.3 is devoted to the literature on the gear teeth numbers. For the sake of completeness previous works on strength design, torsional analysis and inertia of gear boxes are briefed in Section 2.4.

2.2 KINEMATIC ARRANGEMENT OF GEARBOX

Koenigsberger [1] studied on graphical solutions of kinematical arrangement diagrams. He reported the study of Germar, which covered the possibilities of gear drives from 4 to 18 steps by graphical methods. He stated that Germar was the first, who introduced kinematical arrangement and speed diagrams. In selection of most suitable kinematical arrangement diagram some recommendations are also given.

He also discussed the advantages of geometric progression in stepped drives.

Acherkan [2] explains the kinematical relationships related to arrangement of gears in gearboxes. Analytical and graphical methods are presented. Additional to uniform structures, non-uniform structures are also discussed to increase the range ratio of gearbox. Cutting speed relations, advantages of geometric progression and standard progression ratios are described. Recommendations related to kinematical arrangement diagrams are also considered.

Akün [3] and Bodur [4] reviewed the analytical studies on kinematical arrangement diagrams of gearboxes .

Sanger [5] studied kinematic arrangement diagrams for three shaft gear trains not having more than 5 transmissions in a group. He presented equations for maximum range ratio of output speeds, maximum progression ratio and limits on the maximum and minimum transmission ratios.

White [6] set the equations to calculate gear diameters in terms of smallest gear diameter and progression ratio for 9-speed gear box using 3 shaft with 9 gears. He set the formulas based on a preselected kinematic arrangement diagram. It is, also suggested to use 10-gears to obtain progression ratio between speed steps.

White and Sanger [7,8,9] derived the equations to obtain gear diameters with known values of minimum transmission ratio and progression ratio for 9-speed and 4-speed gear trains using 3 shafts. Again the equations are based on a selected kinematical arrangement diagram. They showed the effects of minimum transmission ratio and progression ratio on the size of gearbox .

Bush, Osman and Sankar [10] obtained the equations for the diameters of gears for general kinematic arrangement diagram of 9 speed 3 shaft gearbox . The equations can be equated to get single composite or double composite arrangements as the previous studies. They prepared a computer program to solve the general diameter equations and fed the relevant data from the kinematic arrangement diagram and found the diameters according to the given constraints.

2.3. GEAR TEETH NUMBERS

A detailed study on the determination of gear teeth number is presented by Chironis [11]. He mentioned about 'five ways to find gear ratios', a paper prepared by McComb and Matson. These methods are mentioned below.

1. Logarithm Method

This method utilizes a table of gear ratio logarithms, which makes it unsuitable for computer programming.

2. Smithson Conjugate Fractions Method

Smithson firstly introduced the method in 1952. Conjugate fractions are used to find gear teeth numbers. Two pairs of gears are said to conjugate, if the difference of their cross product is unity. For a gear ratio

$$\frac{a}{b} \times \frac{c}{d}$$

it is conjugate if

$$a \cdot d - b \cdot c = \mp 1 \qquad (2.1)$$

If they form a conjugate fraction, it is also true that $\frac{a + c}{b + d}$ gives a gear ratio between a/b and c/d.

Smithson Conjugate fraction method approaches to the required gear ratio more rapidly than Gray Method, but to start with the method teeth numbers of pairs a/b and c/d must be known. Also for calculated gear ratios, integer teeth numbers must be found by factorization of numbers to form the gear set.

3. Rappaport Algebraic Method

In this method non-linear equations with four unknowns, representing four gears are reduced to linear equations with two unknowns, solution of these algebraic equations give the gear teeth numbers.

4. Gray Calculator Method

This method is the simplest of all the other methods. Simple arithmetic is used to find integer teeth numbers. The method is explained in detail in Section 3.8.1. and most suitable for computer programming.

5. McComb Matson Calculator Method

To use this method table of equivalent factors in gear ratios is required. Method needs least trial and error, but not suitable for computer programming.

Another method to find gear teeth numbers is introduced by Orthwein [12]. He used continued fraction method to find gear teeth numbers for a given gear ratio.

Reddy [13] has given the equations to find gear teeth numbers for a specified error range and for a given center distance. The method has used the advantage of corrected gears and acceptable results are achieved. This method is explained in Section 3.8.2. in detail and used in computer program.

2.4. BENDING STRENGTH DESIGN, SURFACE STRENGTH DESIGN, TORSIONAL ANALYSIS AND INERTIA OF GEAR BOXES

Bending strength and surface fatigue strength design are attempted by many researchers. Dudly [14], Shigly [15], Chironis [11], Niemann [16], Merrit [17], Faires [18], Buckingham [19] and Johnson [20]. In this thesis, bending strength and surface fatigue strength design are not considered in detail. It is just tried to find a few points that could be used as constraints during development of kinematical arrangement diagrams

Torsional analysis is again reviewed to find constraints at the very early stage of the design process. Marchelek [21], Hatter [22], and Edward [23] handled torsional analysis. Marchelek gave an equation to calculate the reflected elasticity of a gear train.

Inertia of gear trains is studied by Wright [24] and Selfridge [25]. They gave equations to calculate equivalent inertia of a compound gear train.

CHAPTER 3

KINEMATIC ARRANGEMENT

3.1. INTRODUCTION

The method of obtaining kinematic arrangement diagram of a gearbox is presented in this chapter.

To construct a kinematic arrangement diagram, the following items must be known:

- number of output speeds
- values of output speeds (minimum speed and pregression ratio are also enough)
- number of transmission groups
- minimum transmission ratio of each group
- input motor speed

Number and values of output speeds are determined according to the requirements of machine tool. A specific cutting speed must be used for optimum machining with a given cutting tool. Cutting speed is obtained by multiplication of turning speed and periphery of the workpiece or the cutting tool. So a series of output speeds must be provided by the gearbox.

Section 3.2. presents general considerations and terminology related to gearboxes. The relations between output speeds and cutting speeds are explained in Section 3.3. Standard Values of Progression Ratio and Selection Criteria are given in Section 3.4. Section 3.5. discusses the construction of kinematic arrangement diagram and equations are given. In the next sections, transmission ratios are determined and recommendations are given. The two methods to determine gear teeth numbers are briefed in Section 3.8. Section 3.9. deals with percentage error on spindle speeds. Proportional elasticity and inertia of a geartrain is discussed in Section 3.10.

3.2. GENERAL CONSIDERATIONS

The terminology related to gearboxes are explained below.

SPEED - turning speed or output speed of a gearbox.

PROGRESSION RATIO - ratio of two successive output speeds.

KINEMATIC ARRANGEMENT DIAGRAM - diagram representing the arrangements of gears in a gearbox.

SPEED RANGE RATIO - is the ratio of maximum output speed to minimum output speed.

GROUP - a number of transmissions between any two shafts. Number of groups is equal to the number of shafts minus one.

CHARACTERISTIC OF A GROUP - power of pregression ratio between two succeding transmission ratios in the group.

SPEED CHART - a similar diagram to kinematic arrangement diagram which shows the actual values of input, output and shaft speeds.

Kinematical arrangement of gears in a gearbox can be shown graphically on a diagram, which is called kinematic arrangement diagram or structural diagram. The kinematic arrangement diagram contains the data, which are namely: number of transmission groups, number of transmissions in each group, the relative order of the groups in the train of transmissions, characteristic of each group and the relation between transmission ratios.

The kinematic arrangement diagram is constructed by drawing horizontal parallel lines at equal distances from each other to represent the shafts. The speeds are plotted horizontally on a logarithmic scale on the shafts.

The last shaft represents the spindle and speeds are drawn at equal distances given by equation:

$$\frac{n_j}{n_{j-1}} = \varphi$$

and

$$\log(n_j) - \log(n_{j-1}) = \log\varphi \qquad (3.1)$$

The transmission ratios between two axes are indicated by horizontal distances between the corresponding speed values. If the distances between axes are equal, the slope of the lines joining the speed values on different axes are an indication of the corresponding gear ratios.

A gearbox with 3 shafts will have 2 groups. Output speeds are obtained by engagements of gear pairs in two groups. To obtain 6 speeds, one of the groups must provide 2 transmissions and the other 3 transmissions, no matter which one is the first or second group, the possible arrangements of gears on the shafts are shown in Fig. 3.1. b and Fig. 3.2.b. Figure 3.1.a and Fig. 3.2.a are the representations of these arrangements. These figures are simple to construct and they also represent the transmission ratios. These diagrams are called kinematic arrangement diagram of a gearbox.

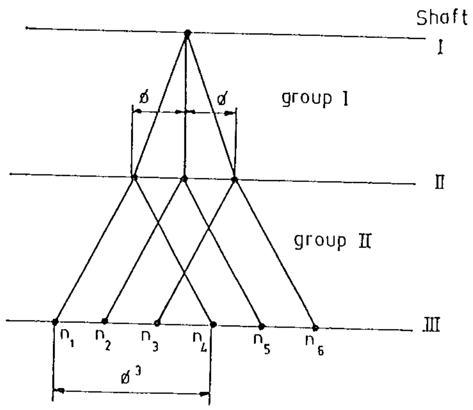


Figure 3.1.a Kinematic Arrangement Diagram of 6 Speed 3 Shafts Gearbox

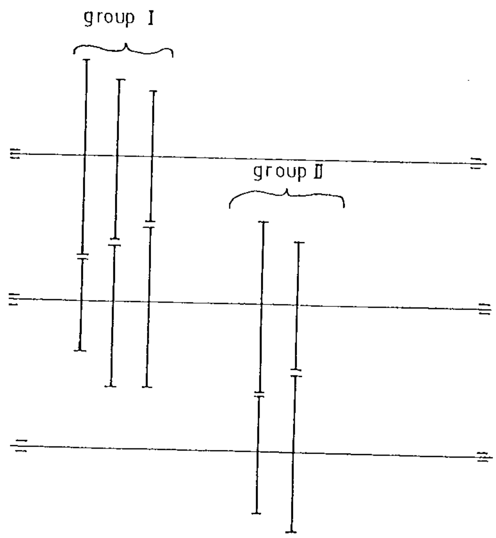


Figure 3.1.b Arrangement of Gears in the same Gearbox

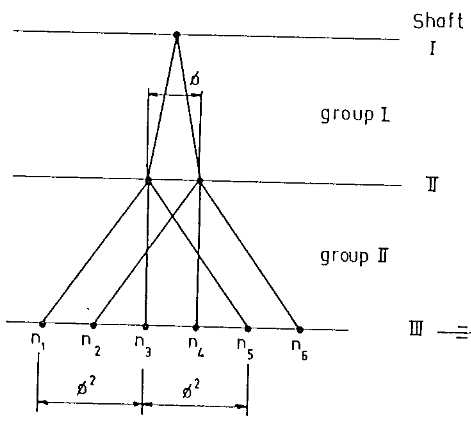


Figure 3.2.a Kinematic Arrangement Diagram of 6 Speed 3 Shafts Gearbox

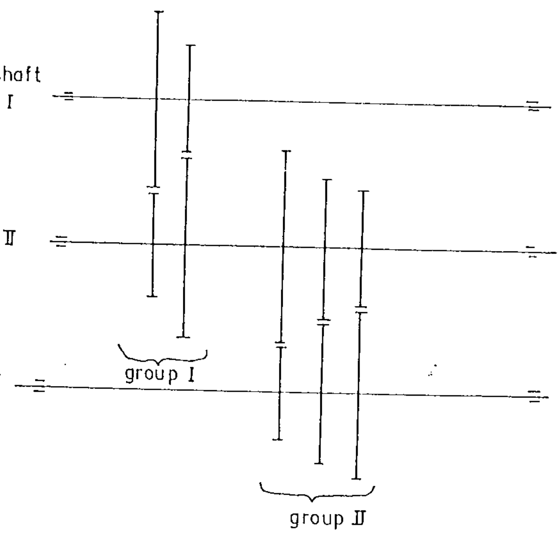


Figure 3.2.b Arrangement of Gears in the same Gearbox

As it is seen, there are two possibilities to obtain a 6 speed gearbox. One is with 3 transmissions in the first group and 2 transmissions in the second group. The other is obtained with 2 transmissions in the first group and 3 transmissions in the second group.

What about the distances between the transmissions in the same group and the other groups? As it will be explained in detail, later in this chapter; the ratio of succeding two transmissions is constant in a group and represented by φ^{A}, A being an integer. If the two figures are carefully examined, it can be seen that (Fig. 3.1.a) in the first group the ratio equal to φ and in the second group, it is equal to φ^{3}. In Fig. 3.2.a, however, it is φ and φ^{2} for groups 1 and 2 respectively. In both figures, the characteristics of the first groups are 1 and characteristics of second groups are equal to 3 in Fig. 3.1.a and 2 in Fig. 3.2.a. These two diagrams have the same order of arrangements, because the first group is treated as the main group with a characteristic of 1 and the second group is the first extension group in both diagram. Any group in the arrangement might be main group, first extension group; second extension group or etc. Order of these groups is called the order of arrangement in the gearbox kinematic arrangement diagrams.
So why not to have second group as the main group and first group as the first extension group. This will yield another two dissimilar kinematical arrangements of a 6 speed gearbox as shown in Figures 3.3. and 3.4

As a result, the number of transmissions can be changed between groups giving an (r!) arrangements and order of arrangement of groups also gives another (r!) possibilities for a gearbox with r groups.

Hence, there should be $(r!)^{2}$ possible kinematical arrangements which will necessitate computerized work for $r > 4$.

With the computer program, all of these (r!) possibilities can be obtained and also similar arrangements can easily be eliminated.

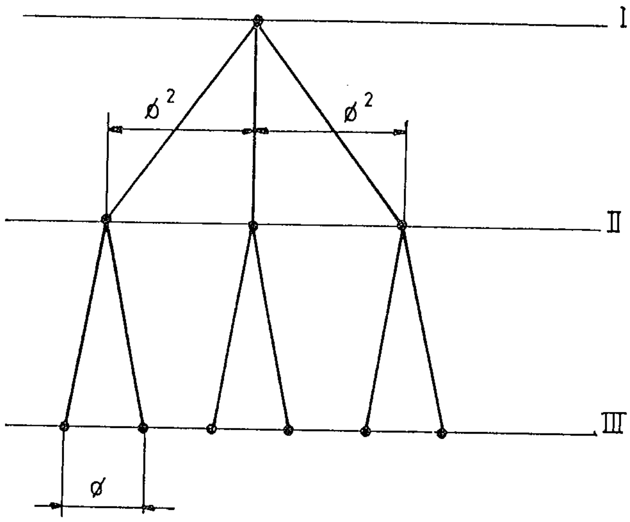


Figure3.3 Kinematic Arrangement Daigram with Second Group as the Main

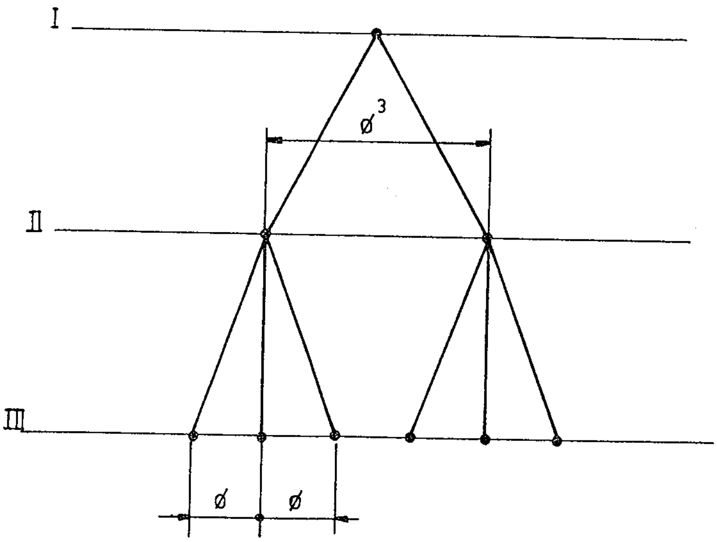


Figure 3.4 Kinematic Arrangement Diagram with Second Group as the Main Group

3.3. CUTTING SPEED REQUIREMENTS

On single purpose machine tools only one cutting speed and feed rate are used. But general purpose machine tools have to provide a range of speeds to obtain optimum machining. According to the cutting tool and required surface quality different cutting speeds are needed.

The cutting action is obtained by two different movements. On a turning machine , it is the rotation of workpiece and linear movement of cutting tool. On a milling machine, it is the rotation of cutting tool and linear movement of workpiece. And on a planing machine, it is both linear movements. Diameter is considered to mean workpiece or cutting tool diameter according to the type of machine tool.

Cutting speed is expressed as :

$$\vartheta = \pi d n \qquad (3.2)$$

and rotational speed (for simplicity will be called speed from here on) is calculated as:

$$n = \frac{\vartheta}{\pi d} \qquad (3.3)$$

Minimum and maximum speeds are given by the equations:

$$n_{min} = \frac{\vartheta_{min}}{\pi \; d_{max}} \qquad (3.4)$$

$$n_{max} = \frac{\vartheta_{max}}{\pi \; d_{min}} \qquad (3.5)$$

Speed range ratio is defined as the ratio of maximum speed to minimum speed.

$$R_n = \frac{n_{max}}{m_{min}} \qquad (3.6)$$

$$R_n = \frac{\vartheta_{max}}{d_{min}} \cdot \frac{d_{max}}{\vartheta_{min}} = \frac{d_{max}}{d_{min}} \cdot \frac{\vartheta_{max}}{\vartheta_{min}} \qquad (3.7)$$

$$R_n = R_d \cdot R_\vartheta \qquad (3.8)$$

Rd is the range of diameters expressed as :

$$R_d = \frac{d_{max}}{d_{min}} \qquad (3.9)$$

and Rv is the range of cutting speeds given by

$$R_\vartheta = \frac{\vartheta_{max}}{\vartheta_{min}} \qquad (3.10)$$

For linear cutting movements speed range ratio depends only upon the range of cutting speeds. It depends both upon the speed range and diameter range for circular cutting movements.

If the cutting speed for a workpiece diameter falls outside the range of cutting speeds, it can not be machined economically because either the machining time will be longer or tool life will be smaller than the expected values. By using stepless drive optimum cutting speeds and feed rates can be provided for machining different diameters. But Acherkan [2] states that in majority of cases, modern machine tools are still designed with a stepped series of speeds. In these stepped speed drives, the optimum cutting speed can be obtained under certain conditions. It is essential to establish maximum and minimum cutting speeds and the ratio between the speeds.

The relation between the cutting speeds and the workpiece diameter is linear for any spindle speed. According to the cutting tool used and workpiece material to be machined, maximum and minimum cutting speeds are determined and plotted on the speed diameter diagram (See Fig. 3.5)

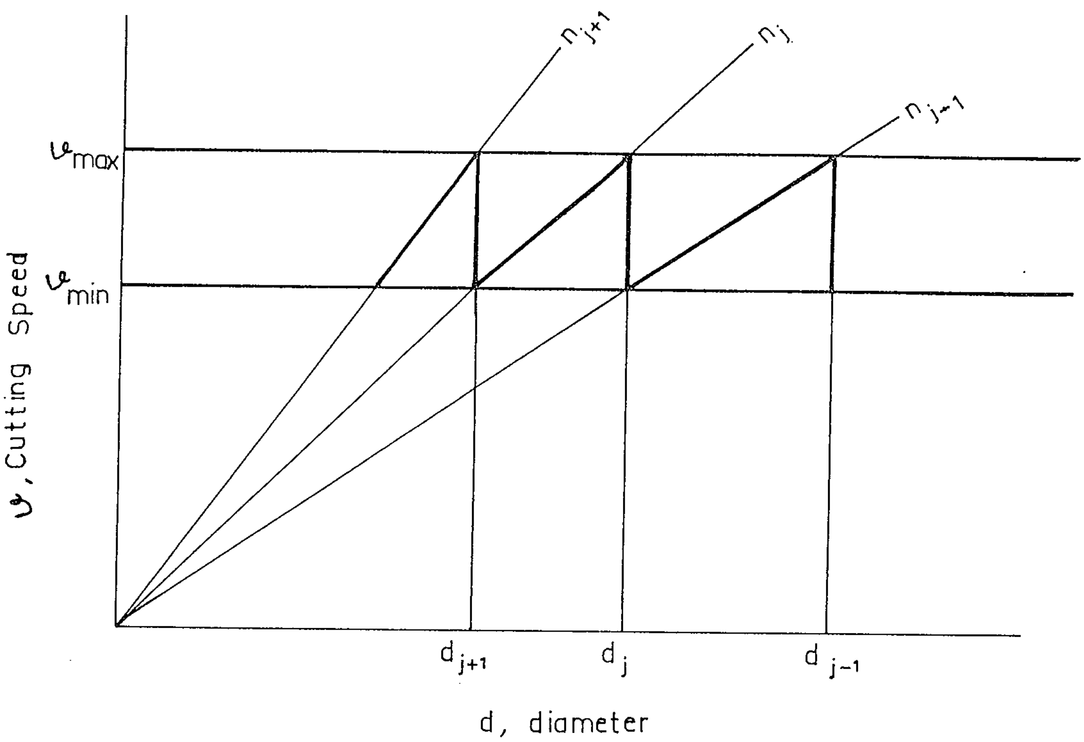


Figure 3.5 Speed Diameter Diagram

This diagram is also called Saw Tooth Diagram

The workpiece diameter must be between dj and dj+1 for the spindle speed nj. If other diameters are machined with spindle speed nj, either the machining time is increased or the tool life is shortened; because obtained cutting speed falls outside the cutting speed range. And spindle speed nj-1 is used for the diameter between dj-1 and dj.

In designing a machine tool, the range of diameters and cutting speed range must be selected according to the purpose of machine tool. The methods of obtaining stepped spindle speeds are discussed in the following sections.

3.3.1. ARITHMETIC PROGRESSION METHOD

A constant progression number is used between two succeeding spindle speeds. This system uses a constant upper limit and a decreasing lower limit with increasing workpiece diameter for cutting speeds (Fig. 3.6)

$$\begin{aligned} n_2 - n_1 &= \varphi & n_2 - n_1 &= \varphi \\ n_3 - n_2 &= \varphi \rightarrow & n_3 - n_1 &= 2\varphi \\ &\vdots & & \\ n_z - n_{z-1} &= \varphi \rightarrow & n_z - n_1 &= (z-1)\varphi \end{aligned} \qquad (3.11)$$

Where φ is the progression number, which is equal to :

$$\varphi = \frac{n_z - n_1}{z-1} \qquad (3.12)$$

Where N_z = n max and n_1 = n min and z is the total number of stepped speeds.

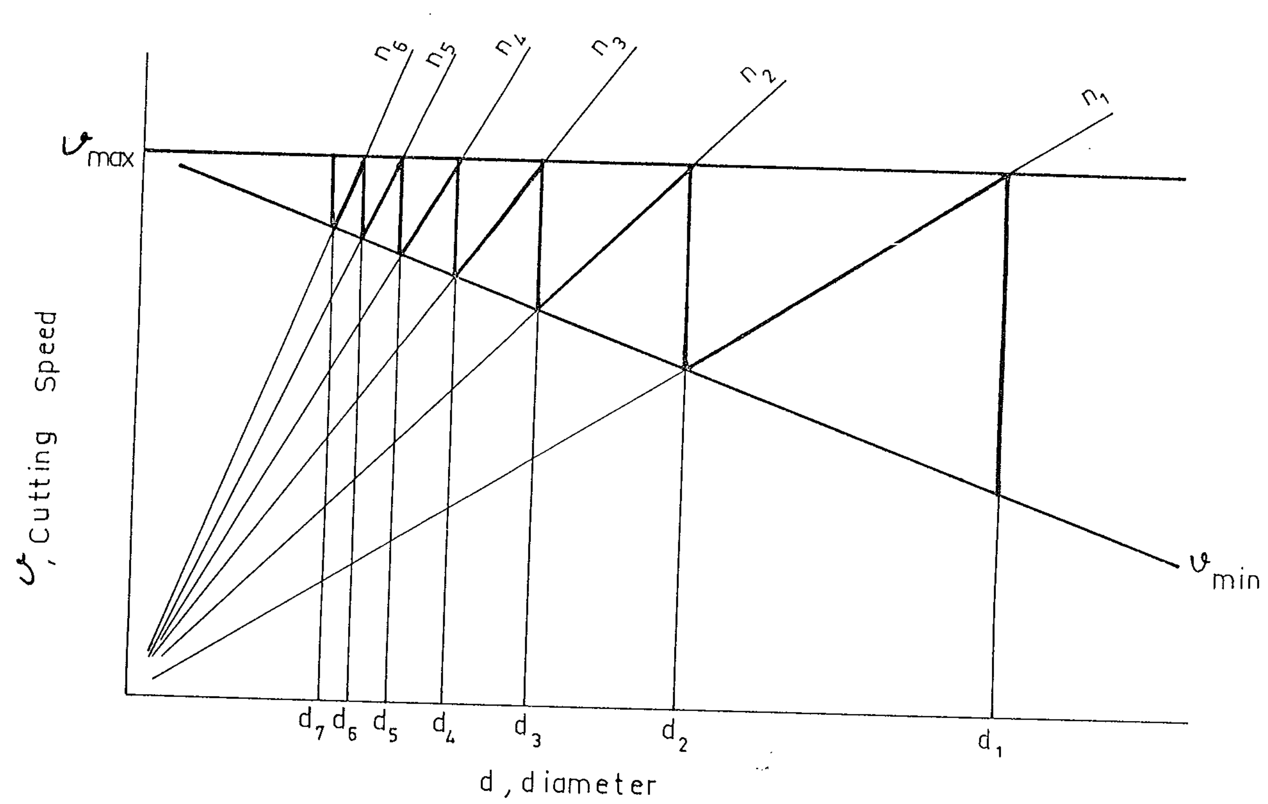


Figure 3.6 Speed Diameter Diagram for Arithmetic Progression

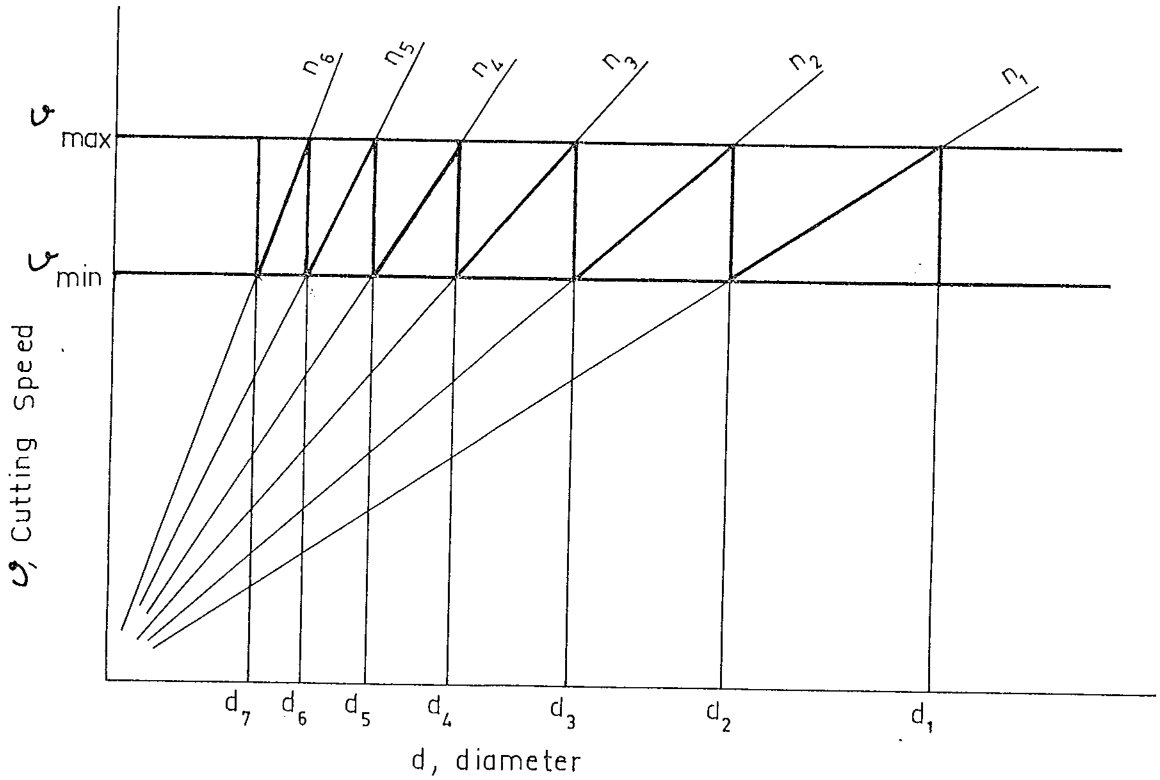


Figure 3.7 Speed Diameter Diagram for Geometric Progression

Speed range ratio is equal to :

$$R_n = \frac{n_z}{n_1} = 1 + \frac{\varphi(z-1)}{n_1} \qquad (3.13)$$

As it is seen on the diagram (Fig. 3.6) speed steps are concentrated on the small diameter region. There are less speed steps for the larger diameters. If such a speed steps are used on a machine tool, larger diameters can not be machined effectively because optimum cutting speeds can not be obtained.

3.3.2. GEOMETRIC PROGRESSION METHOD

A constant progression ratio is used between two consecutive spindle speeds in this system. The following relations can be written between the diameters, spindle speeds and the cutting speeds.

$$n_1 = \frac{\vartheta_{max}}{\pi\ d_1} = \frac{\vartheta_{min}}{\pi\ d_2}$$

$$n_2 = \frac{\vartheta_{max}}{\pi\ d_2} = \frac{\vartheta_{min}}{\pi\ d_3}$$

$$n_{z-1} = \frac{\vartheta_{max}}{\pi d_{z-1}} = \frac{\vartheta_{min}}{\pi\ d_z}$$

$$n_z = \frac{\varphi_{max}}{\pi\ d_z} = \frac{\varphi_{min}}{\pi\ d_{z+1}} \qquad (3.14)$$

A constant ratio is obtained between two succeeding spindle steps

$$\frac{n_z}{n_{z-1}} = \frac{\vartheta_{max}}{\vartheta_{min}} = \varphi \qquad (3.15)$$

Constant maximum and minimum cutting speeds are used for all diameters in this system (Fig. 3.7)

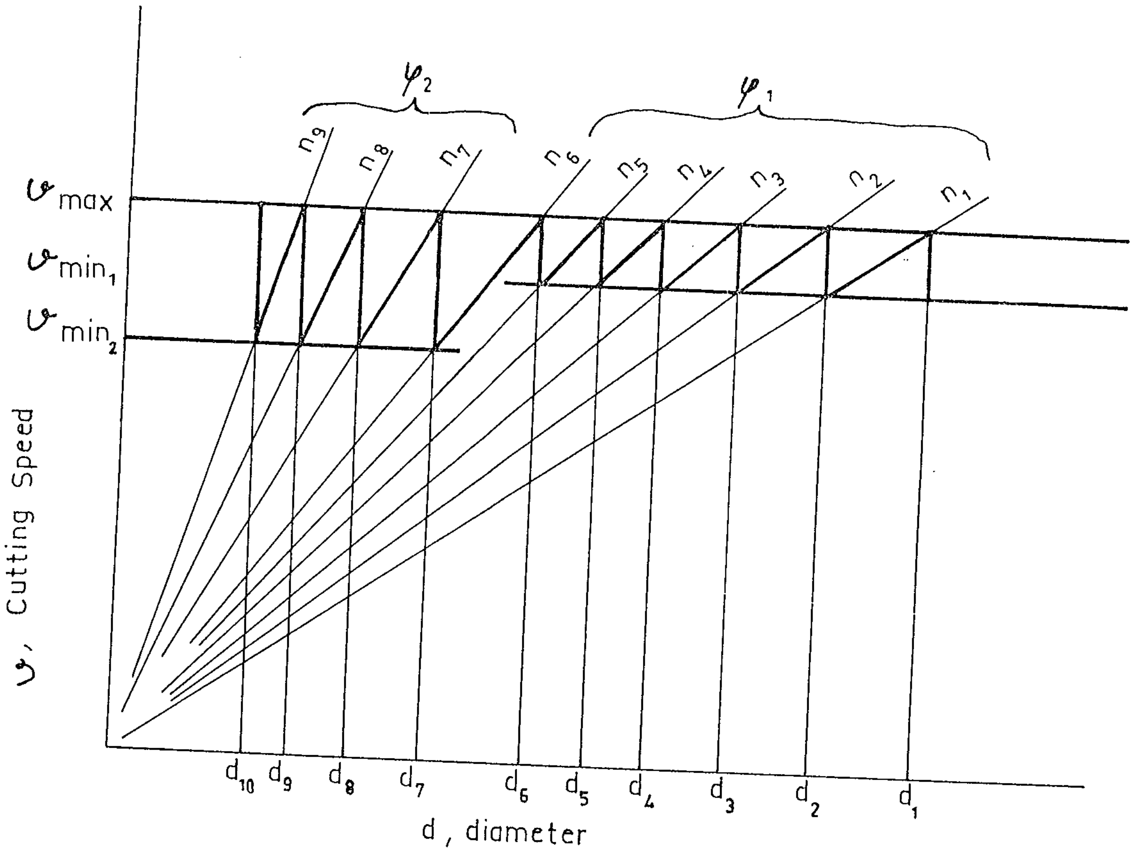


Figure 3.8 Modified Geometric Progression

Spindle speeds can also be calculated with the following formula

$$n_2 = n_1 \varphi$$

$$n_3 = n_2 . \varphi = n_1 \varphi^2$$

$$n_z = n_{z-1} . \varphi = n_1 \varphi^{z-1} \qquad (3.16)$$

And speed range ratio is obtained as

$$R_n = \frac{n_z}{n_1} = \varphi^{z-1} \tag{3.17}$$

Constant progression ratio is calculated with

$$\varphi = \sqrt[z-1]{\frac{n_z}{n_1}} = \sqrt[z-1]{R_n} \tag{3.18}$$

Geometrical progression can be further improved by using combination of two geometric progression series. Smaller geometric progression ratio is used for larger diameters. For example, in Fig. 3.8 smaller progression ratio φ_1 is used for diameters between d_1 and d_7 and a greater progression ratio, φ_2 is used for smaller diameters than d_7. With this modification, more speed steps are obtained for larger diameters. Also smaller progression ratio can be used for any diameter range which the machine tool is mostly expected to work with or to machine.

Geometric progression series provides an even distribution of output speeds in the range of speeds. It is most widely used for machine tool gearboxes and accepted as the most advantageous method [1].

Some of the advantages of geometric progression are discussed below:

1. From a geometric progression series with the ratio φ, if x-1 members of the group are excluded in every succeeding x member subgroup, a new geometric progression series is obtained with the ratio of φ^x.

2. If each element of a geometric progression series is multiplied with a factor of φ^y, the whole series is shifted by y members.

3. If the elements of a geometric progression series are multiplied with a constant "c", a new geométric progression series is obtained with the same progression ratio, but having an initial member "c" times greater.

3.4. STANDARD PROGRESSION RATIOS

As it is mentioned standard spindle speeds are based on the preferred numbers, which are standardized by DIN 804 and presented in Table 3.1.

Preffered numbers are established as decimal geometrical series. Preffered numbers are used because they give the optimum distribution in an interval. Decimal geometrical series have the properties of both decimal numbers and geometrical progression. The series are established by dividing geometrically each decimal group (1-10,

Table 3.1 STANDARD SPINDLE SPEEDS UNDER LOAD ACCORDING TO DIN 804

BASIC RANGE	RANGE R 20/2	R A N G E R 20/ 3			RANGE R 20 / 4		R A N G E 20 / 6		
					(1400)	(2800)			
φ=1.12	φ=1.25	φ =1.4			φ = 1.6		φ = 2		
100				1000					
112	112	11.2				112	11.2		
125			125						
140	140			1400	140				1400
160		16							
180	180		180			180		180	
200				2000					
224	224	22.4			224		22.4		
250			250						
280	280			2800		280			2800
315		31.5							
355	355		355		355			355	
400				4000					
450	450	45				450	45		
500			500						
560	560			5600	560				5600
630		63							
710	710		710			710		710	
800				8000					
900	900	90			900		90		
1000			1000						

10-100, 100-1000).

Progression ratio in decimal geometrical series is expressed by equation (3.19). R being the range of numbers is always equal to 10.

$$\varphi = \sqrt[z-1]{R} = \sqrt[E]{10} \tag{3.19}$$

E being a whole number is selected to be 40, 20, 10 or 5. With these assumptions, progression ratios are shown in Table 3.2.

φ_{40} is the finest series and containes all the other series. By changing initial value different series can be obtained.

In selection of a standard progression ratio, electric motor speeds must also be taken into account. In practical applications used motor speeds are: 3000-1500-1000-750-600-500-375-300.

Speeds of direct current motors can be selected as required, but to be in correspondence with a.c. motors generally used speeds are: 3000-2000-1500-1200-1000-750-600-500-375-300.

These speeds are off-load speeds. On load, they are reduced by 6.5% and 13.5% according to the quality and production method of motor [3]. Thus the motor speeds will be lowered to : 2800-1800-1400-1120-900-710-560-450-355-280. These speeds are all contained in R20 and R20/2 series and partially contained in the other series. Motor speed is generally given as the starting point of geometric progression series.

Considerations of motor speeds require that progression ratio is equal to: [2]

$$\varphi = \sqrt[E]{2} \tag{3.20}$$

Thus standard values of progression ratio must satisfy both equations (3.19) and (3.20).

$$\varphi = \sqrt[E_1]{2} = \sqrt[E_2]{10} \tag{3.21}$$

These two equations are satisfied , if

$$E_1 = 3E' \quad \text{and} \quad E_2 = 10E' \qquad (3.22)$$

Where E_1 and E_2 are whole numbers. With the addition of values $\varphi = \sqrt{2} = 1.41$, $\varphi = \sqrt[1]{2} = 2$ and $\varphi = \sqrt[4]{10} = 1.78$, the standard values of progression ratios are completed and given in Table 3.2.

TABLE 3.2 STANDARD PROGRESSION RATIOS

φ	1.06	1.12	1.26	1.41	1.58	1.78	2
$\sqrt[E]{2}$	$\sqrt[12]{2}$	$\sqrt[6]{2}$	$\sqrt[3]{2}$	$\sqrt{2}$	$\sqrt[1.5]{2}$	$\sqrt[1.2]{2}$	$\sqrt[1]{2}$
$\sqrt[E]{10}$	$\sqrt[40]{10}$	$\sqrt[20]{10}$	$\sqrt[10]{10}$	$\sqrt[20/3]{10}$	$\sqrt[5]{10}$	$\sqrt[4]{10}$	$\sqrt[20/6]{10}$
A_{max}	5	10	20	30	40	45	50

Maximum relative percentage loss in cutting speed is expressed by Acherkan [2] as

$$A_{max} = \frac{\varphi - 1}{\varphi} \times 100 \qquad (3.23)$$

and depends only on φ. For each series, its values are also given in Table 3.2.

3.4.1. SELECTION OF PROGRESSION RATIO AND NUMBER OF SPEED STEPS

When the maximum and minimum spindle speeds are selected for the machine tool being designed, the range ratio Rn = is fixed. number of speed steps or progression ratio must be decided. With the selection of one, the other is determined. It can easily be seen from Fig. (3.9) that number of speed steps increases rapidly with a decrease in progression ratio.

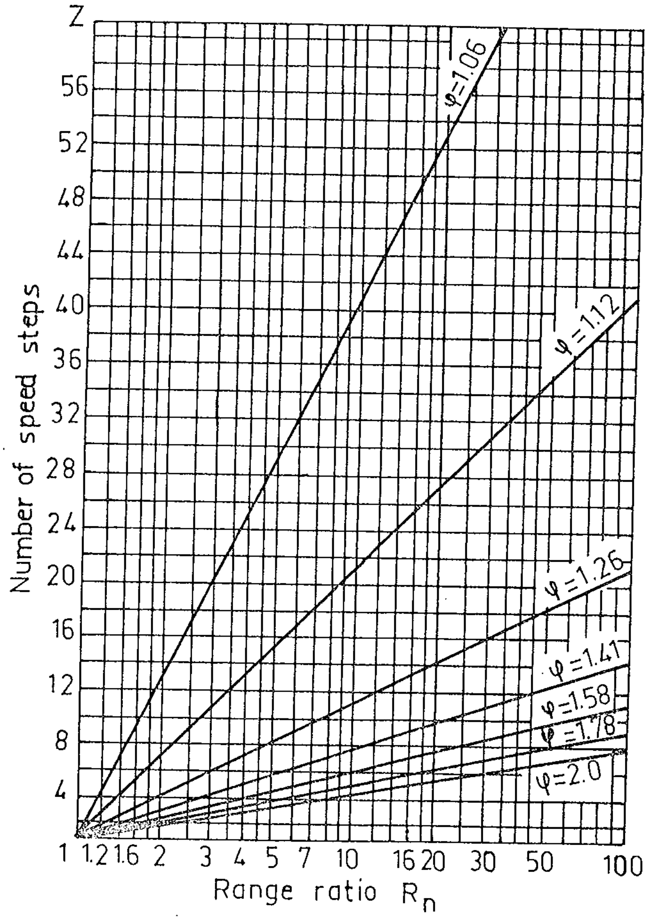


Figure 3.9 Relationships between Speed Range Ratio, Number of Speed Steps and Progression ratio

Progression ratio must be selected in such a way that for a given speed steps, it has to provide the speed range. If a smaller progression ratio is selected for a given speed range, number of speed steps is increased. This will increase the cost of gearbox, for final decision, the following items should be taken into consideration [2]:

1. Progression ratios φ= 1.26 and φ=1.41 are widely used for general purpose machine tools and give quite satisfactory operation.

2. Progression ratios φ= 1.12 and φ=1.26 proved to be satisfactory if used with change gears in the drive gear-train for automatic and semi-automatic machine tools for mass production.

3. Large progression ratios φ=1.58 and sometimes φ=1.78 are used in small machine tools for machining small work diameters.

4. For heavy machine tools smaller progression ratios φ=1.26, φ=1.12 and sometimes φ=1.06 are used .

5. It is a good practice to select a number of speed steps Z which has factors of 2 and 3. So that

$$Z = 2^{E_1} . 3^{E_2} \qquad (3.24)$$

where E_1 and E_2 are whole numbers. The most frequently used values are:

$$Z = 3, 4, 6, 8, 12, 18 \text{ and } 24$$

The range ratio R_n and number of speed steps Z can vary between quite large limits. According to the purpose of machine tool, the nature of manufacturing process, the type of cutting tool to be used and especially versatility of new machine tool these values must be specified. If the machine tool is more versatile and different kinds of cutting tools are to be used, the range ratio must be larger for efficient operation. For example, in cylindrical grinding machines, wheel spindle speed range ratio is $Rn \leqslant 2$ for wheel diameters varying within the limits $Rd \leqslant 2$. On the other hand for a horizontal boring machine, the range ratio of feed drive reaches $Rs \cong 1000$ in certain models.

For machine tools with a rotary cutting motion, number of speed steps is taken as $Z \leqslant 36$.

The values of Rn and Z should be much smaller in special machine tools, than in general purpose machine tools.

3.5. KINEMATIC ARRANGEMENT DIAGRAMS

The spindle drive gearbox should provide required number of speed steps, with a given geometrical progression ratio to give the required series of speeds.

Spindle speeds can be obtained either by simple gear trains on two shafts or with compound gear trains. With simple gear trains, which is also called single transmission, any spindle speed can be obtained by engagement of proper gear pair. But if compound gear trains are used which composes more than one transmission group, consecutive engagement of gear pairs in each group must be provided to obtain any spindle speed. Fig. 3.10 gives arrangement of gears and its simple representation for an 18-speed gearbox.

As it is mentioned before, because of its advantages, geometric progression is used with compound gear trains in machine tools. For the construction of kinematical arrangement diagrams, the items mentioned in Section 3.2 are discussed in detail in following sub-sections.

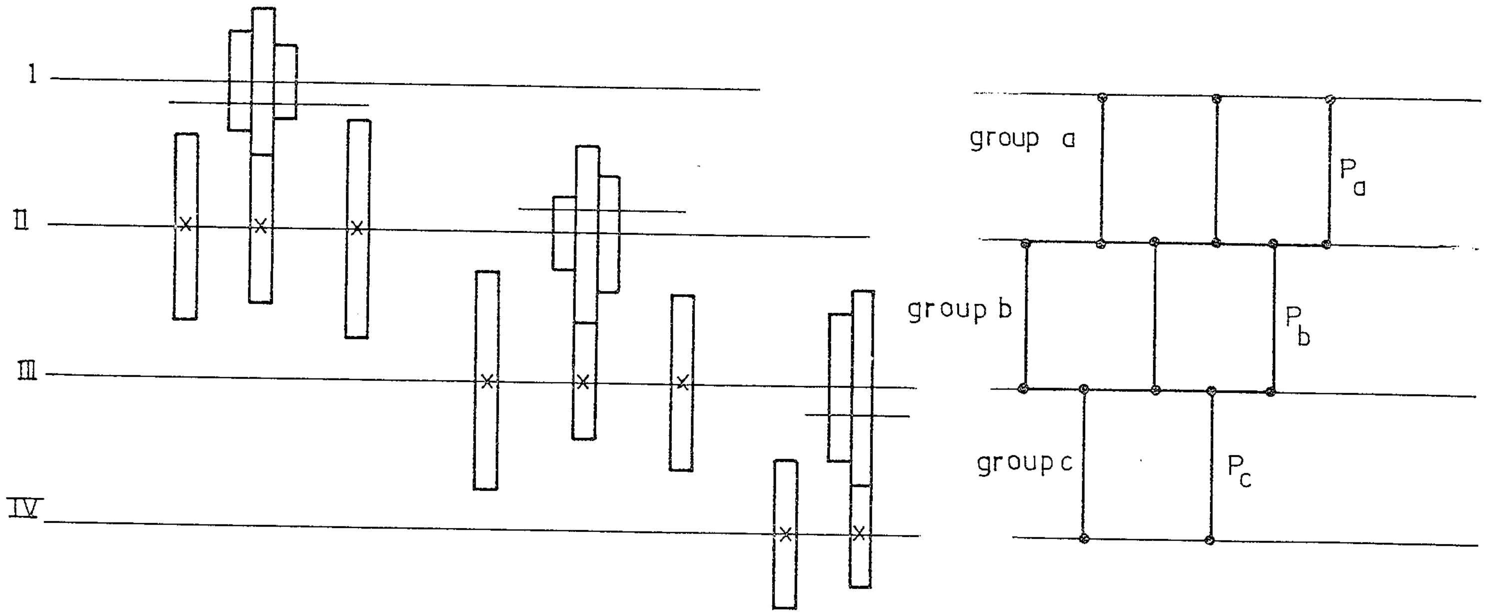


Figure 3.10 Arrangement of Gears of an 18 Speed 4 Shaft Gearbox

3.5.1. NUMBER OF SPEED STEPS

The number of spindle speed steps is equal to the multiplication of number of simple trains in each group. A group is the total engagements between any consecutive shafts. Total number of group in a gear box is equal to the total number of shafts minus one. And simple train is just a transmission obtained by a gear pair. Denoting the number of simple trains in each group by Pa, Pb, Pc..Pr (where subindexes a, b, c,...r refers to transmission groups) number of spindle speeds Z is equal to

$$Z = P_a . P_b . P_c \; ..... \; P_r \qquad (3.25)$$

3.5.2. SPEED RANGE RATIO

Total transmission ratio of a compound gear train is equal to the product of transmission ratios of simple train. Thus, maximum and minimum transmission ratios of gear train can be calculated as

$$i_{max} = i_{amax} . i_{bmax} . i_{cmax} ..... i_{rmax} \qquad (3.26)$$

$$i_{min} = i_{amin} . i_{bmin} . i_{cmin} ..... i_{rmin} \qquad (3.27)$$

Range ratio of the whole train is equal to:

$$R_n = \frac{n_{max}}{n_{min}} = \frac{i_{max}}{i_{min}} = R_a . R_b . R_c ..... R_c \qquad (3.28)$$

Where $R_a = \frac{i_{amax}}{i_{amin}}$ and similarly Rb, Rc, . . .Rr are the range ratios of the transmission groups in the train.

3.5.3. BASIC EQUATION FOR KINEMATIC ARRANGEMENT

To obtain a series of speeds use of multiplier transmission groups is an important property of geometrical series of speeds. The conditions to change the speeds of such a geometrical series are determined by the kinematic arrangements of multiplier groups.

The situation can be explained by an example. Assume a gear box having a speed range ratio Rgb is linked with a simple gear train with constant transmission ratio (Fig. 3.11). The gear box provides geometrical series of speeds from ni to nk.

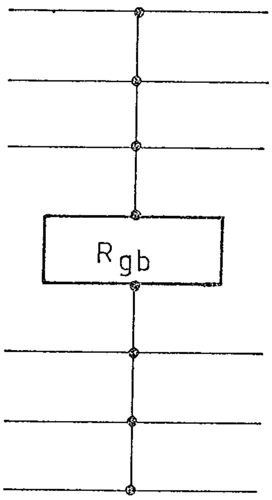


Figure 3.11 Structural Diagram of a Gear Train

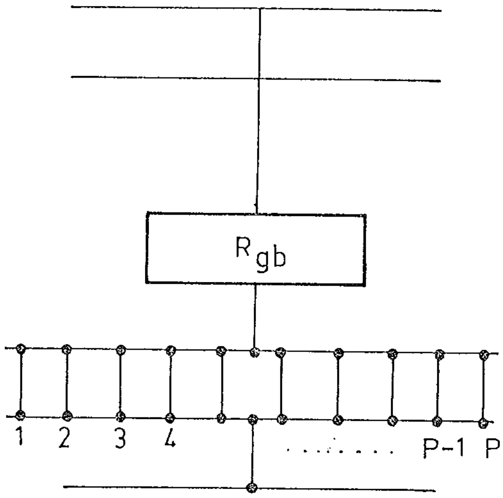


Figure 3.12 Structural Diagram of a Gear Train with a Multiplier Group

Next a multiplier group is introduced to the train with p number of transmissions (Fig. 3.12). The multiplier group has the transmission ratios of i_1, i_2, i_3 ...i_p . With the engagement of first transmission in the multiplier group, a spindle speed series can be obtained as

$$n_1, n_2, n_3 \ldots\ldots\ldots n_{k-1}, n_k$$

The engagement of second transmission gives another spindle speeds series

$$n_{k+1}, \quad n_{k+2}, \quad n_{k+3}, \quad \ldots\ldots n_{2k}$$

The ratio of elements in the second series to the elements in the first series is always equal to $\frac{i_2}{i_1}$.

Hence, $$\frac{i_2}{i_1} = \frac{n_{2k}}{n_k} = \frac{n_{2k-1}}{n_{k-1}} \ldots\ldots = \frac{n_{k+1}}{n_1} = \frac{n_k \cdot \varphi}{n_1} = R_{gb} \cdot \varphi \qquad (3.29)$$

Thus with the engagements of other transmissions in the multiplier group, the similar ratios will be obtained. The following relationships concerning the transmission ratios of the multiplier group can be stated

$$i_1 : i_2 : i_3 : \ldots\ldots : i_p$$

$$= n_1 : n_{k+1} : n_{2k+1} : n_{3k+1} : \ldots\ldots : n_{(p-1)k+1} \qquad (3.30)$$

$$= n_1 : n_1 R_{gb} : n_1 (R_{gb} \cdot \varphi)^2 : n (R_{gb} \cdot \varphi)^3 : \ldots\ldots : n_1 (R_{gb} \cdot \varphi)^{p-1}$$

The transmission ratios in the multiplier group construct a geometrical series with the progression ratio $\varphi_p = \varphi R_{gb}$.

Thus the relationships exist between the transmission ratios

$$i_1 : i_2 : i_3 : \ldots\ldots : i_p = 1 : \varphi R_{gb} : (\varphi R_{gb})^2 : \ldots\ldots : (\varphi R_{gb})^{p-1} \qquad (3.31)$$

Where Rgb is the speed range ratio of the whole transmission preceding the multiplier group. Since each group is a multiplier group of the whole transmission kinematically preceding it, the above equation is true for all the groups which build up the gear box.

3.5.4. CHARACTERISTIC OF A TRANSMISSION GROUP

The progression ratio of a transmission group can be expressed simply as

$$\varphi_p = R_{gb}\varphi = \varphi^{z_{k-1}} \cdot \varphi = \varphi^{z_k} = \varphi^x \qquad (3.32)$$

Where z k is the total number of speed steps kinematically preceding the given group which is equal to the characteristic of the group.

The general equation (3.31) can be expressed as follows for any transmission group.

$$i_1 : i_2 : i_3 : \ldots : i_p = 1 : \varphi^x : \varphi^{2x} : \ldots : \varphi^{(p-1)x} \qquad (3.33)$$

In the kinematic arrangement of simple transmissions the main group is the first group in the whole complex with

$$x_1 = z_k = 1$$

The second transmission group, -so called first extension group- has the characteristic of $X_2 = P_1$
where P_1 is the number of transmissions in the main group.

In the third group -the second extension group- the characteristic is equal to $X_3 = P_1 . P_2$ where P_2 is the number of transmissions in the first extension group. A general equation can be laid down to find the characteristic of any group in the gear box.

$$x_r = p_1 \cdot p_2 \ldots p_{r-1} \qquad (3.34)$$

In applying these equations to a whole transmission complex, it has to be decided which group to be the main group, first extension group or second extension group etc. Equation (3.25) is also called the structural formula which the order of a,b,c, ... r subindexes give the main first extension, second extension etc. groups. So knowing the number of transmissions in each group, order of arrangement of groups and characteristics of the groups kinematical arrangement diagram can be constructed and all the possible solutions are obtained.

3.6 DETERMINATION OF TRANSMISSION RATIOS

As explained in the previous sections, with given initial data kinematical arrangement diagrams are constructed. Briefly, number of speed steps is factored out to determine number of transmission in each group. Order of transmission groups in the drive is decided and characteristic of each group is calculated with Eq.(3.34). Now kinematic arrangement diagram can be established and used for future calculatios.

All the transmission ratios are expressed in terms of progression ratio and minimum transmission ratio. Not to cause excessively large diameters of gears minimum and maximum transmission ratios are limited. These limits are accepted in all practical applications. They are ¼ for minimum transmission ratio and 2/1 for maximum transmission ratio for spur gears used in machine tools.

$$i_{minlim} = \frac{1}{4} \quad \text{and} \quad i_{maxlim} = \frac{2}{1} \qquad (3.35)$$

If helical gearing is used maximum transmission ratio limit is increased to 2.5/1. And sometimes maximum limiting ratio of 4/1 is used in small machine tools.

Generally accepted range of transmission ratios for feed gear boxes is $1/5 \leqslant i \leqslant 2.8/1$.

The limiting range ratio is thus, established for a two shaft transmission group as

$$R_{lim} = \frac{i_{maxlim}}{i_{minlim}} = 8 \qquad (3.36)$$

The minimum transmission ratio i min for the whole drive is calculated and expressed in the form of exponent of the progression ratio of the series. Thus,

$$i_{min} = \frac{n_{min}}{n_i} = \frac{1}{\varphi^E} \qquad (3.37)$$

Exponent E is taken from standard series of numbers to be used in machine tools. Minimum transmission ratio of each group is determined in such a manner that their product is equal to i min (3.27). They can be expressed in the form $i = \varphi^{\mp u}$, so that algebraic sum of exponents U is equal to E

A general kinematic arrangement diagram is given in Fig. 3.13 and explained below

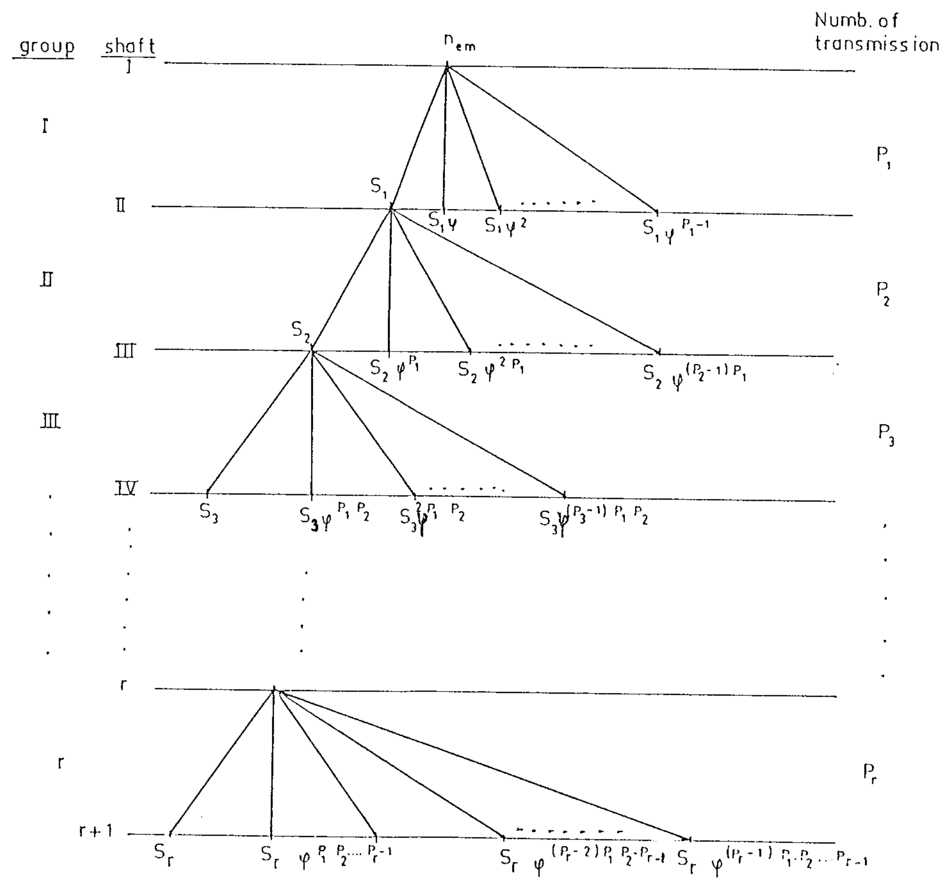


Figure 3.13 General Kinematic Arrangement Diagram

The minimum transmission ratios of the groups are expressed with letter S for simplicity, i.e.

$$S = i_{minlim}$$

$$S_1 = i_{minlim_1}$$

$$S_2 = i_{minlim_2} \qquad (3.38)$$

$$S_r = i_{minlimr}$$

$$S = S_1 . S_2 . S_3 ..... S_r$$

The main group consisting of p_1 transmissions has a characteristic of $x_1 = 1$ The transmission ratios in the main group are calculated to be

$$S_1,\ S_1\varphi,\ S_1\varphi^2,\ S_1\varphi^3,\ ....,\ S_1\varphi^{p_1-1} \qquad (3.39)$$

In the second group, the number of transmissions is p_1 and the characteristic of the group is equal to $x_2 = p_1$. The transmission ratios of the group are

$$S_2,\ S_2\varphi^{p_1},\ S_2\varphi^{2p_1},\ S_2\varphi^{3p_1},\ ....,\ S_2\varphi^{(p_2-1)p_1} \qquad (3.40)$$

And by the same way the transmission ratios of the third group are found to be

$$S_3,\ S_3\varphi^{p_1p_2},\ S_3\varphi^{2p_1p_2},\ S_3\varphi^{3p_1p_2},\ ....,\ S_3\varphi^{(p_3-1)p_1.p_2} \qquad (3.41)$$

The transmission ratios of the following groups can be found similarly up to r'th group. Transmission ratios of the r'th group:

$$S_r,\ S_r\varphi^{p_1 p_2 \cdot\cdot p_{r-1}},\ S_r\varphi^{2p_1 p_2 \cdot\cdot p_{r-1}},\ \ldots\ldots,\ S_r\varphi^{(p_r-1)p_1 p_2 \cdot\cdot p_{r-1}} \tag{3.42}$$

When these equations are set up, they can be arranged in any order to obtain the same output speeds from the whole complex, and number of transmissions in each group can be changed on the same kinematic arrangement of a stepped gear box. For a specified number of transmission groups r and a specified number of transmissions in each group there will be numerous design options. For the structural diagram there will be r! solutions, i.e.

$$Z = p_1 \cdot p_2 \cdot p_3 \cdots p_r = p_2 \cdot p_1 \cdot p_3 \cdots p_r = p_3 \cdot p_2 \cdot p_1 \cdots p_r \tag{3.43}$$

If there are m groups with an equal number of transmissions in each, the number of design options will be reduced to $\frac{r!}{m!}$

Arrangement order of the main, first, second .... groups also introduces r ! option for each kinematic diagram.

Each group may be considered as main group, first, second, or any extension group up to the last one. Therefore, total number of options

$$\frac{(r!)^2}{m!} \tag{3.44}$$

will be obtained. For example for the structure Z = 18 = 2x3x3 in which r = 3, and m = 2, the number of design options will be $\frac{(3!)^2}{2!} = 18$

3.7. ANALYSIS AND RECOMMENDATIONS ON KINEMATIC ARRANGEMENT DIAGRAM

For a constant transmitted power, transmitted torque indirectly proportional to the speed of the shaft (Eq. 3.45) if speed is decreased the torque will increase which needs a greater shaft diameter and a greater face width or module of the gear is required. For gear boxes used in machine tools which covering a range of speed steps, certain gears have to transmit considerably higher torques. Due to higher tooth loads, these gears have to be larger than the others and require more space.

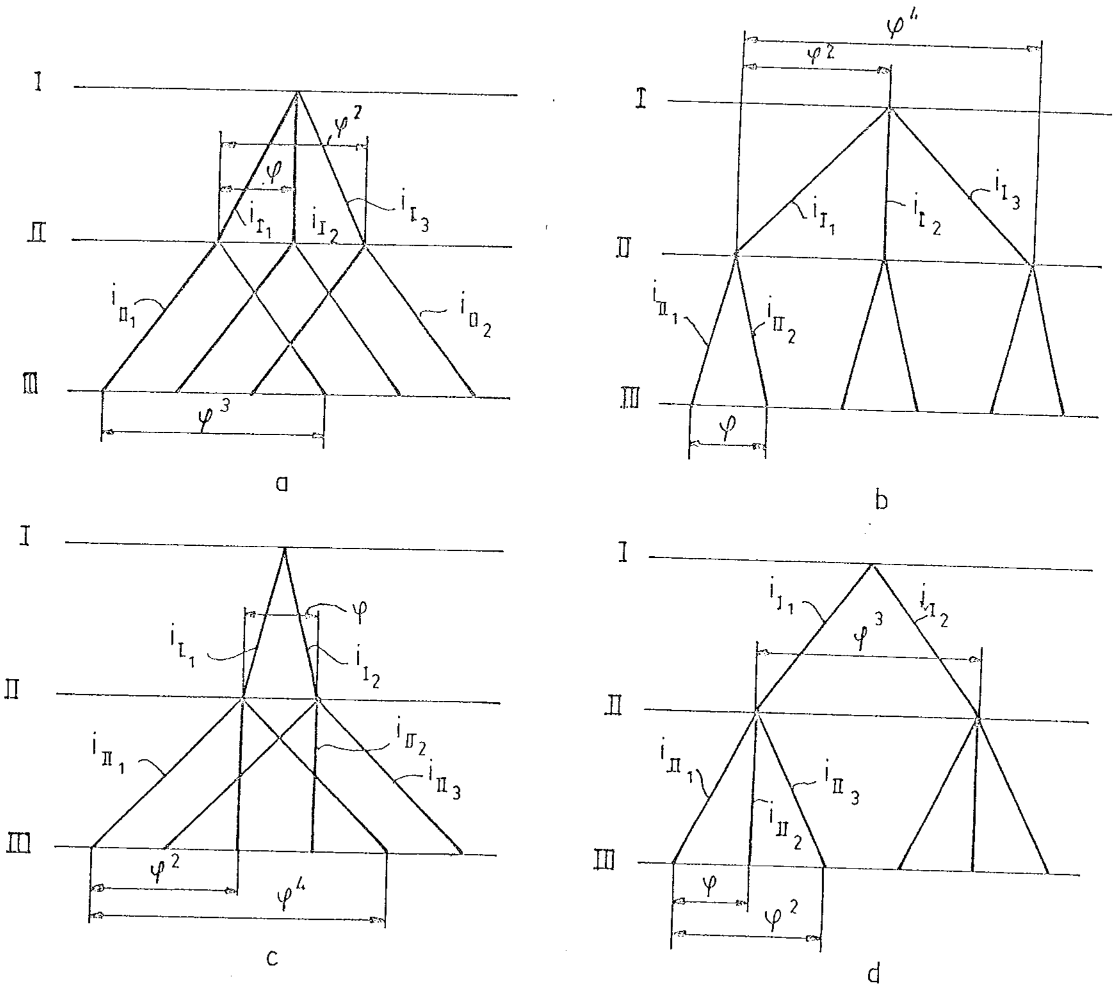


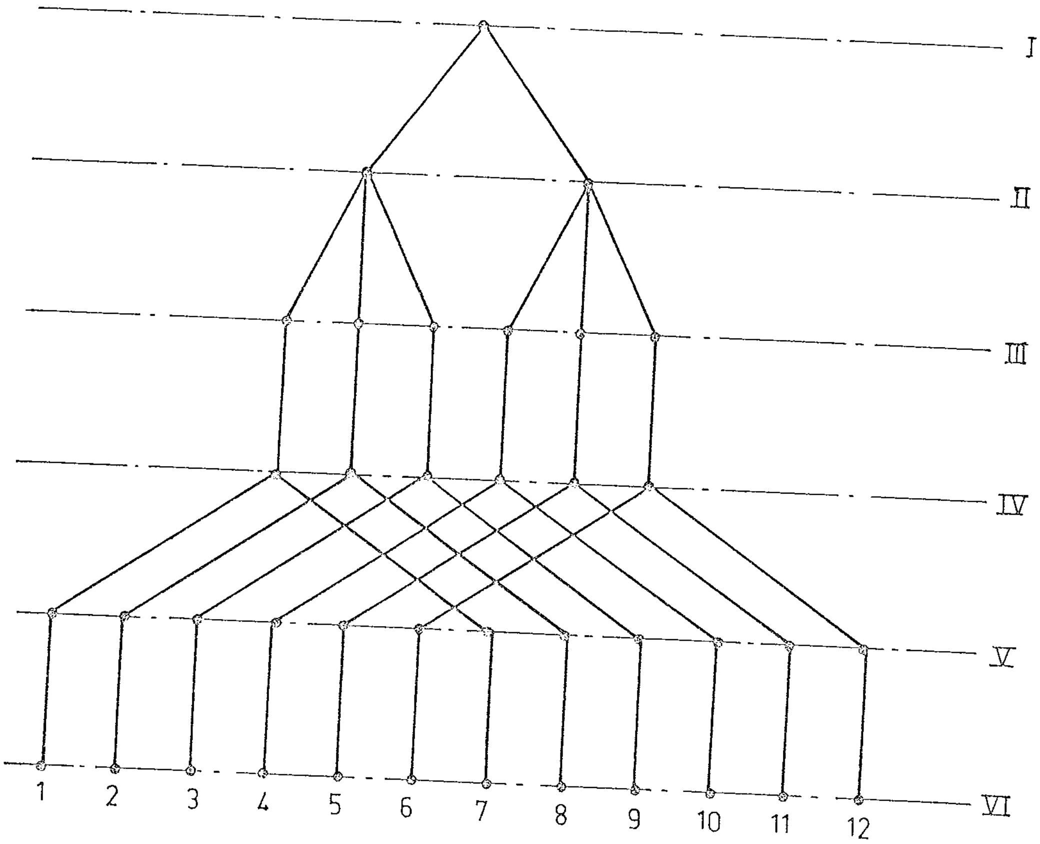
Figure 3.14 4 Kinematic Arrangement Diagrams of 6 Speed 3 Shaft Gearbox

Let us consider the gear box with three shafts and 6 speeds as discussed in Section 3.2. Figures (3.1a), (3.2a), (3.4) and (3.5) are re-drawn in Fig. 3.14 to follow the discussions below easily.

In the case of layout Fig. 3.14 a, the highest transmission ratios occur in the second group. So that only the gears transmitting the ratio i_{II1} and producing three lowest speeds, have to be stronger. In Fig. 3.14 b, the two lowest speeds are transmitted by two sets of gears, with transmission ratios i_{I1} and i_{II1}. For this layout, the highest transmission ratio, i.e. the greatest speed change occurs in the first group. Therefore the gears for reduction i_{I1} must also be stronger. Similar situations occur also for the arrangements in Fig. 3.14 c and d.

Another point is the numerical values of transmission ratios. These transmission ratios and ranges should not exceed the limiting values mentioned in Section 3.6. In the arrangements a and c , largest transmission range occur in the second group with φ^3 and φ^4 respectively. It occurs in the first groups with values of φ^4 and φ^3 respectively in arrangements b and d. It is stated by Koenigsberger [1] that it is also preferable to have the largest transmission range in the last group for reasons of tooth strength and space requirements. This means that theoretically, optimum arrangement is that shown in Fig. 3.14 a which requires the smallest transmission range.

In addition to the kinematic arrangement diagram speed charts are constructed almost by the same way. The kinematic arrangement diagram indicates the relationships between the transmission ratios of the group transmissions but does not give the actual values of ratios and speeds. Values of transmission ratios for all the transmissions in the drive and speeds of all the shafts are determined by constructing the speed chart. Kinematic arrangement diagram and speed chart of Ajax Lathe are shown in Figures (3.15) and (3.16).



Figures 3.15 Kinematic Arrangement Diagram the Gearbox used for Main Drive of Ajax Lathe

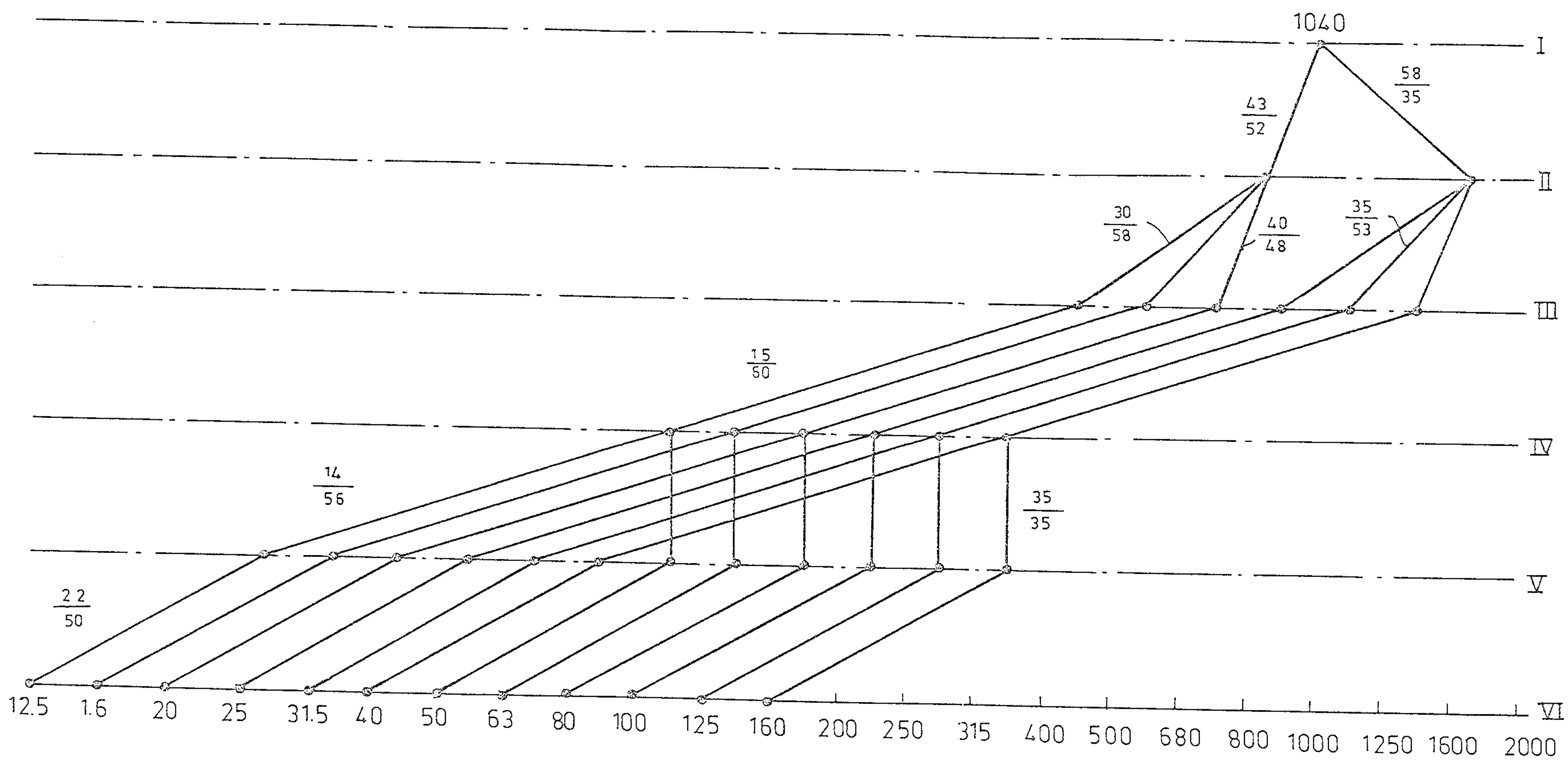


Figure 3.16 Speed chart of the Gearbox used for Main Drive of Ajax Lathe

3.7.1. WEIGHT OF THE DRIVE

The size of the gears in the transmissions of a gearbox increase with an increase in the transmitted torque. The torque is equal to

$$T = 716 \frac{h_p}{n} \eta \qquad (3.45)$$

By taking logarithm, it is found

$$\log T + \log n = \log 716 + \log h_p + \log \eta \qquad (3.46)$$

This equation is represented in [2] on a logarithmic graph (Fig. 3.17)

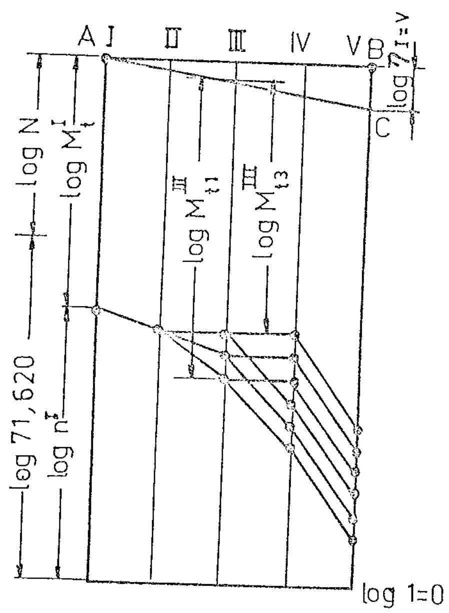


Figure 3.17 Transmitted Torque at each Spindle Speed

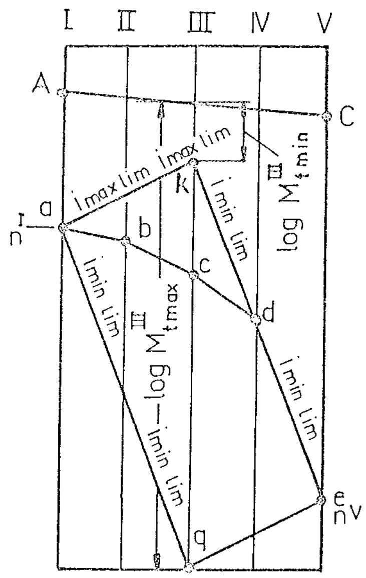


Figure 3.18 Transmitted Torque for Different Minimum Transmission Ratio of Each Group

The horizontal line AB represents the value of (log 716 + log hp) with the assumption of $\eta = 1$. If the efficiency of transmission groups are same, line AC shows the transmitted power with losses. That graph can be used to find the value of transmitted torque at any speed. The vertical distance between line AC and speed point on the shaft is equal to log T. In Fig. 3.17, maximum and minimum transmitted torque is shown for the third shaft.

Fig. 3.18 is similar to Fig. 3.17, but only minimum transmission ratios are shown with different values. Again the distance between line AC and speed point represents the torque developed. 3 values for minimum transmission ratios are shown on the figure. Broken line a k e represents the situation for using maximum limiting transmission ratios for the first and second groups and minimum limiting transmission ratios for the third and fourth groups. For this option minimum transmitted torques are obtained for the intermediate shafts. Even the weight may be at a minimum it is stated [2] that group transmissions can not be used for the first two links. For the option of using minimum limiting transmission ratios for the first and second groups and using maximum limiting transmission ratios for the third and fourth groups develops maximum torques on the intermediate shafts and is not advantageous. Therefore it is preferable to follow the broken lines abcde in designing a reduction train. The transmission ratios are reduced to a smaller and smaller value as the train approaches the spindle. And the transmission ratio between spindle and preceeding shaft is taken equal to minimum limiting valve.

To reduce the weight of the drive, the number of transmissions in the groups must decrease along the train from the electric motor to the spindle. Thus, if

$$Z = p_a \cdot p_b \cdot p_c \cdots p_r \qquad (3.25)$$

It is advisable to apply

$$p_a > p_b > p_c > \;\ldots\ldots\; > p_r \qquad (3.47)$$

Simple transmissions should be arranged nearer to the spindle. For a total number of transmissions, this arrangement ensures a larger number of transmissions with less weight and less transmissions with heavier components, because the design torque increases as the train approaches to the spindle.

3.7.2. MINIMUM NUMBER OF TRANSMISSIONS

For a required number of speed steps
the number of speed steps in the transmissions are determined with equation (3.25) Total number of transmission ,

$$S_p = p_a + p_b + p_c + \;\ldots\; + p_r$$

will be minimum, if

$$p_a = p_b = p_c = \ldots = p_r = \sqrt[r]{Z} = p \qquad (3.48)$$

if the number of transmission groups, r, is not specified, then minimum number of transmissions will be obtained with $p = 2$ or $p = 3$. These are actually the numbers of transmissions that is used for sliding gears [2]. Total number of gears used is twice the number of transmissions. But with change gears two different transmissions are obtained with one pair of gears.

3.7.3. MINIMUM NUMBER OF TRANSMISSION GROUPS

Minimum number of transmission groups is obtained if the range ratio of each group is equal for a specified range ratio Rn.

$$R_n = R_a . R_b . R_c \;.....\; R_r \qquad (3.28)$$

$$R_a = R_b = R_c = \;.... = R_r \qquad (3.49)$$

Gear boxes with uniform groups having p=2 or p=3 transmissions in each group provide the minimum number of transmissions in all the groups but at the same time, the maximumnumber of groups are required. Thus,to reduce the number of transmissions, number of groups must be increased or vice versa.

For long reduction trains in heavy machine tools uniform groups with minimum number of transmissions is used. But for a short reduction gear train in high speed machine tool minimum number of groups is preferred.

3.7.4. CHARACTERISTIC OF GROUPS

The most advantageous kinematic arrangement of a gear drive is obtained with increasing characteristic of the groups from electric motor to the spindle. With the structural equation

$$Z = p_1 . p_2 . p_3 . \;....\; p_r \qquad (3.25)$$

The characteristic of the groups must be

$$x_1 < x_2 < x_3 < \ \ldots\ldots \ < x_r \qquad (3.50)$$

When this order of group characteristics are used maximum speeds of the intermediate shafts will be smaller for the same minimum speeds. Ackerkan [2] states that if these order of characteristics are used, the manufacturing accuracy of components will be less, dynamic loads, vibrations, wear of components, friction losses will be reduced and the efficiency is increased.

3.8 DETERMINATION OF GEAR TEETH NUMBERS

In the following two subsections Gray Method to find gear teeth numbers for a given gear ratio within the limits is discussed and the method is explained to find gear teeth numbers for a given gear ratio with a constant center distance.

Computer program uses Gray method to find gear teeth numbers for minimum transmission ratio of each group. The other method is used to find other gear teeth numbers in the same group for fixed center distance.

3.8.1. GRAY METHOD

This method is first introduced by Gray in 1953 and his study is presented in [5]. Gray Method is the simplest to find gear teeth numbers. Given gear ratio is multiplied with integer numbers and the product is divided by the same integer number. That is

$$R_f = \frac{x \cdot R}{x} \tag{3.51}$$

x being an integer and R is the required gear ratio. The multiplication of xR is carried out until it approaches an integer number within specified error limits. Although the method is tedious for hand calculations, it work effeciently with computer and gives acceptable results.

For examples for a gear ratio of 0.82692, it is found

$\frac{19}{23} = 0.826087$ with -0.100737 % error

and for a ratio of 0.25, it is easily found $\frac{18}{72} = 0.25$ with no error.

3.8.2. GEAR TEETH NUMBER CALCULATIONS FOR A FIXED CENTER DISTANCE

In design of a gear box when the kinematic arrangement diagram is constructed, gear ratios in each group can be found on the diagram. In construction of kinematic arrangement diagram, at the first stage gear ratios and gear teeth numbers are found for the minimum transmission ratios in each group. When these teeth numbers are calculated, the center distance, which is proportional to the teeth number of driven and driving gears is fixed. The teeth numbers of other gears in the same group must also have the same center distance with the required error range. Reddy [13] presentd a method for calculation of gear teeth numbers for a fixed center distance. The advantage of corrected gears are taken into account. By this way, total teeth number representing the center distance can be changed within the limits of ± 3 teeth. Figure (3.19) shows two pairs of gears with the same center distance. Gear teeth numbers are represented by A, B, C, D..

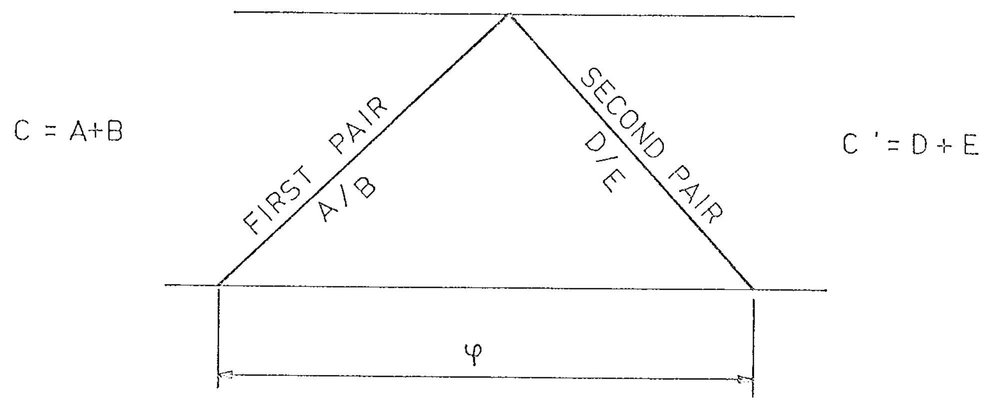


Figure 3.19 Two Gear Pairs Representing a Group

By use of corrected gears new center distance, c' will be equal to one of the followings;

$$
\begin{aligned}
c' &= c \\
c' &= c \pm 1 \\
c' &= c \pm 2 \qquad (3.52) \\
c' &= c \pm 3
\end{aligned}
$$

With the progressio ratio φ, the ratio of second to first transmissions ratio is equal to,

$$\frac{D/E}{A/B} = \varphi = \frac{DB}{AE} \qquad (3.53)$$

$$C = A + B$$
$$C - A = B$$

$$\frac{C - A}{A} = \frac{B}{A}$$

$$(\frac{C}{A} - 1) = \frac{B}{A} \qquad (3.54)$$

and

$$C' = D + E$$
$$C' - D = E$$
$$\frac{C' - D}{D} = \frac{E}{D}$$

$$(\frac{C'}{D} - 1) = \frac{E}{D} \qquad (3.55)$$

Substitution of equations (3.54) and (3.55) into equation (3.53) gives

$$\varphi = (\frac{C}{A} - 1) / (\frac{C'}{D} - 1)$$

$$\frac{C}{A} - 1 = \varphi \ (\frac{C'}{D} - 1)$$

$$[(\frac{C}{A} - 1) / \varphi] + 1 = \frac{C'}{D}$$

$$D = \frac{C'}{[(\frac{C}{A} - 1)/\varphi] + 1} \qquad (3.56)$$

Differentiating equation (3.56) and writing in error form, it is found

$$\frac{\Delta D}{D} = \frac{(\frac{C}{A} - 1)}{(\frac{C}{A} - 1 + \varphi)} \cdot \frac{\Delta\varphi}{\varphi} \qquad (3.57)$$

With an assumption of φ, gear teeth numbers can be found. Assume 1% error on φ, $\frac{\Delta\varphi}{\varphi}$ equals 0.01.

$$D_{max} = D(1 + \frac{\Delta D}{D})$$
$$D_{min} = D(1 - \frac{\Delta D}{D}) \qquad (3.58)$$

For the specified error limit, gear teeth number must lie between D max and D min, if no integer number exists between D max and D min, no gear pair can be found for the value of φ within the error limit.

With the use of equations (3.57) and (3.58) for a given gear ratio, specified center distance and error range gear teeth numbers of second and other gear pairs can be found. This method is used in the computer program to find gear teeth numbers of transmissions except the minimum transmission ratio in all the groups and satisfactory results are obtained.

3.9 ERROR ON SPINDLE SPEEDS

Because of the errors introduced on the gear ratio with the calculated teeth numbers, the spindle speeds will have an error. The total error on spindle speed is equal to the errors of gear pairs providing the spindle speed. This error range is also standardized. DIN 804 specifies the limits with ±2%. And ISO/R229 and Indian standards put the deviation +2% and -3%. Any speed can be written as

$$n_w = n_i \prod_{j=1}^{r} \varphi_{jkj} \qquad (k_j = 1,2,3 \ldots p_j) \qquad (3.59)$$

Where nw is equal to any output speed

ni is the input speed

r is the total group number

pj is the total transmission number in group j

The value of φ_{jk_j} is calculated to be (3.33)

$$\varphi_{jkj} = \varphi^{x_j(k_j-1)} \tag{3.60}$$

With the calculations based on gear teeth number φ_{jkj} will contain an error $\Delta\varphi_{jkj}$.

Taking the logarithms of equation (3.59)

$$\log nw = \log n_1 + \sum_{j=1}^{r} \log \varphi_{jkj} \quad (k_j=1,2,3,\ldots.p_j) \tag{3.61}$$

Differentiating and writing in error form, it is obtained

$$\frac{\Delta n_W}{n_W} = \sum_{j=1}^{r} \frac{\Delta\varphi_{jkj}}{\varphi_{jkj}} \quad (k_j = 1,2,3,\ldots.p_j) \tag{3.62}$$

Denoting the percentage error on spindle speed and percentage error on gear ratio by

$$\varepsilon_{nw} = \frac{\Delta_{nw}}{nw} \times 100$$

$$\varepsilon_{jkj} = \frac{\Delta\varphi_{jkj}}{\varphi_{jkj}} \times 100 \tag{3.63}$$

It is found that,

$$\varepsilon_{nw} = \sum_{j=1}^{r} \varepsilon_{jkj} \quad (k_j = 1,2,3 \ldots. p_j) \tag{3.64}$$

The individual errors on gear ratio will have a maximum and minimum value in each group, such that

$$(\varepsilon_{jkj})_{max} > 0$$

$$(\varepsilon_{jkj})_{min} < 0 \tag{3.65}$$

Thus total percentage error on spindle speed will have maximum minimum values

$$\varepsilon_{nwmax} = (\sum_{j=1}^{r} \varepsilon_{jkj})_{max} = \sum_{j=1}^{r} (\varepsilon_{jkj})_{max} > 0$$

and

$$\varepsilon_{nwmin} = (\sum_{j=1}^{r} \varepsilon_{jkj})_{min} = \sum_{j=}^{r} (\varepsilon_{jkj})_{min} < 0 \qquad (3.66)$$

Hence, total range of percentage error in spindle speeds will be equal to ($\varepsilon_{nw\,max}$ - $\varepsilon_{nw\,min}$). This error range should be less than 5% for ISO/R229 and less than 4% for DIN 804. This range of error is calculated and printed out at the end of results of each option in the computer program.

3.10 DYNAMIC PROPERTIES

Any gear train must be checked against torsional natural frequencies. Torsional natural frequencies depend on the stiffness and inertia of each component included in the gear train. Gears are only one of these components. These problems can be overcomed by using a number of different techniques but considerable time is required. Especially at the stage of kinematic arrangement design or selection, it is hard to reach a solution. The following two sections deal with finding a proportional value of elasticity and inertia of a gear box during kinematical analysis, which can be used as determining factors in comparison of options.

3.10.1 ELASTICITY OF GEAR BOX

Marchelek [21] mentiond about a criteria to determine the torsional elasticity of a gear train. Dynamic properties of a gear box are determined mainly by its torsional stiffness. By establishing an equivalent system, these properties can be analysed. Torsional stiffness can be described as the resistance against deflection under a torsional moment. Torsional stiffness coefficient is defined by

$$K = \frac{T}{\theta} \qquad (3.67)$$

Reflected values of stiffness coefficient with respect to any speed is obtained by dividing this coefficient with the square of the ratio of shaft speed to the speed of reflected shaft.

$$K_{js} = \frac{k_j}{i_{jm}^2} \qquad (3.68)$$

$$i_{jm} = \frac{n_j}{n_m} = \frac{\text{shaft speed}}{\text{motor speed}} \qquad (3.69)$$

Elasticity is defined as the reciprocal of stiffness.

$$e = \frac{\theta}{T} \qquad (3.70)$$

and similarly reflected elasticity is calculated as

$$e_{jm} = (e_j)(i_{jm}^2) \qquad (3.71)$$

Total coefficient of torsional ~~stiffness~~ elasticity for a gear box can be written as

$$e_t = \sum_{j=1}^{j=n} e_{jm} = \sum_{j=1}^{j=n} (e_j)(i_{jm})^2 \qquad (3.72)$$

The smaller the value of product of e_j and $(i_{jm})^2$, the total reflected elasticity will be the smaller. Smaller the elasticity coefficient, smaller will be the torsional deflection for a given torsional moment.

The elasticities of elements used in machine tool drives are given by [21]. For a toothed gear, it is equal to

$$e = K \frac{1}{bD^2 \cos^2\alpha} \quad \text{rad/kg-cm} \qquad (3.73)$$

where $K = 24 \times 10^{-6}$ cm²/kg for steel spur gears

b - width of gear, cm

D - outside diameter of gear , cm

d - pressure angle, degree

Outside diameter is related to module and number of teeth by

$$D = m(N+2) \tag{3.74}$$

The face width of gear is also proportional to the outside diameter (10)

$$b \sim d = m(N+2) \tag{3.75}$$

By assuming a 20° pressure angle and a unit module , the equation (3.73) becomes:

$$e \sim \frac{K}{m^3(N+2)^3 \mathrm{Cos}^2\ 20}$$

$$e \sim \frac{2.71793 \times 10^{-5}}{(N+2)^3} \tag{3.76}$$

So the total reflected elasticity will be proportional to

$$e_t \sim 2.71793 \times 10^{-5} \sum_{j=1}^{j=n} \frac{i_{jm^2}}{(N+2)^3} \tag{3.77}$$

This proportional reflected elasticity is calculated for each spindle speed in computer program and can be used as a determining factor in the selection of options.

Bush [10] in his paper says that another measure of dynamic characteristic is the sum of the largest gear ratios in each mesh

$$D = \sum_{j=1}^{j=r} (i_{max})_j \tag{3.78}$$

This dynamic criteria is required to be small for an increased static stiffness.

3.10.2 INERTIA OF GEAR BOX

In torsional vibration analysis of rotating systems, rotation of masses about their centres of mass is the concern. Therefore, the polar moment of inertia of the mass is the parameter used instead of the mass. It is, generally, referred as the "inertia" for simplicity. For a given drive torque the acceleration will be maximum when the inertia is minimum. In machine tool drives to obtain a wide frequency response the acceleration must be maximum [24]. It is, therefore, desirable to minimize the inertia.

Equivalent or reflected inertia of a gear box is obtained by multiplying the actual inertia of gear with the square of the ratio of shaft speed to reflected shaft speed.

$$I_r = I_j (i_{jm})^2 \qquad (3.79)$$

i j m is given by equation (3.69)

Inertia of a gear is given by

$$I = \frac{1}{8} m D^2 \qquad (3.80)$$

With the same assumptions as in Section 3.10.1

$$D = M(N+2)$$

$$m = \frac{\pi D^2}{4} \cdot b\rho$$

$$b \sim D = M(N+2)$$

$$m \sim \frac{\pi\rho}{4} \cdot M^2 (N+2)^2 \cdot M(N+2)$$

$$m \sim \frac{\pi\rho}{4} \quad M^3 (N+2)^3 \qquad (3.81)$$

$$I \sim \frac{1}{8} \quad \frac{\pi\rho}{4} M^3 (N+2)^3 . \; M^2 (N+2)^2$$

$$I \sim \frac{\pi\rho}{32} \quad M^5 (N+2)^5 \qquad (3.82)$$

With this formula, a proportional value of reflected inertia can be calculated on the assumption of a unit module

$$I_r \sim (N+2)^5 . (i_{jm})^2 \qquad (3.83)$$

For each spindle speed these reflected inertias are calculated to be used in the selection of gear arrangement diagram.

CHAPTER 4

4.1. INTRODUCTION

The calculations involved in determining the kinematical arrangements of gearboxes are complicated and laborious as shown in the previous chapter.

It is the aim of this chapter to automate the procedure and to make the kinematical arrangement, which satisfy certain requirements, available to the designer in his final decision.

Section 4.2 introduces the features of the computer program prepared for this purpose. In Section 4.3 , the use of the program is illustrated with the examples chosen from existing machine tools available in the market.

4.2. COMPUTER PROGRAM

The computer program consists of main and subprograms. It is interactive and highly user oriented. By giving proper answers to the questions asked by the program, the user is enabled to obtain all possible arrangements of the Gearbox. If at any stage of the process, the user does not know what to do, by typing ?; messages are typed out on the terminal to guide the user in the rest of the process.

The simplified flow charts and description of the program are given in Appendix (4.1).

Upon initiating the program, a message on how to set up input data is typed out on the terminal. The input data to the program contains number of speed steps, either two of the minimum speed, maximum speed or progression ratio, number of total shafts, number of shafts with one transmission and motor speed. Limitations on minimum and maximum transmission ratios on minimum and maximum teeth numbers, percentage error in gear ratio and percentage error in output speed may also be given at this stage.

Once the data is set up, it can be saved for further use or for further minor changes in the requirements of the gear box. At this stage, the user has the opportunity to check the input values, to change the previous input data, to change the limitations on kinematic values of the gearbox .

With the command 'CONT', program controls the input data against unlogical values. If it is found user is asked to correct the data, if not, program continues to perform the following functions:

- calculate probable order of arrangement
- factorize number of speed steps
- equalize factored speed number with the shaft number
- calculate probable speed distribution and eliminate the same ones

After these calculations, minimum transmission ratio of each group is calculated in such a manner that minimum limiting transmission ratio is assigned to the last group and it is increased up to the first group. Minimum transmission ratios as well as the minimum speed calculated according to these ratios are typed out on the terminal for comparison purpose. At this point, the following options appear on the terminal,

RAT - To enter the required minimum transmission ratios of each group from the terminal

POW - To enter the powers of progression ratio for calculations of minimum transmission ratios

CAL - To assume the previously calculated transmission ratios

If entered 'RAT' or 'POW', the user is guided by typing out the calculated values on the terminal. If 'CONT' is used directly without entering the other options, previously stored minimum transmission ratios are taken by the program. With the use of command 'CONT', the minimum transmission ratios are controlled against the values which are out of set limits of minimum and maximum transmission ratios. If values out of limits are found, user asked to change the transmission ratios. If not, teeth numbers for minimum transmission ratios of each group are calculated with Gray Method.

Found teeth numbers with transmission ratios, percentage error and minimum speed are typed out on the terminal to be checked by the user. If the percentage error on minimum speed is not within the limits two options will be available to change gear teeth numbers or transmission ratios,

RAP - To reenter preceeding options RAT or POW

TET - To enter gear teeth numbers for minimum transmission ratio of any group

Command 'CONT' is used to continue. At this stage, the program is ready to provide all of the possible kinematic arrangements. The user is asked to give one of the following answer for printouts

ALL - To get all of the possible arrangements

REC - To get only recommended arrangements

It is left to the wish of the user to store the output data on disk or not. Then the program continues and assumes a speed distribution from the created matrix. According to the speed distribution, the probability matrix for order of arrangements isrearranged to give only different arrangements.

The program assumes the first order of arrangement and is ready to calculate all of the porometers related to kinematic arrangement for all of the possible and different arrangements. The program iterates all possibilities one by one. The results related to the kinematic arrangement of gearbox are typed out on the terminal and stored on disk if stated at the early stage.

According to the results obtained, the user may want some minor changes or modifications in speed distribution or in teeth numbers. Hence, the following options are made available:

ANY - To continue with the same speed distribution by pressing any key

CHN - To assume the next speed distribution from the matrix

TET - To change gear teeth numbers of any transmission in any group

SEE - To type out the preceeding resusits on terminal

EX - To exit from the loop for possible arrangements

By pressing any key on the terminal keyboard, the iteration continues with assuming another order of arrangement for the same speed distribution. By the same way all of the possible order of arrangements are traced out for the same speed distribution. For the next iteration, the program assumes next speed distribution and continues up to the completion of speed distribution matrix.

During the execution of iterations, if the user enters the option 'TET' and then the option 'SEE', the user will have the chance to analyse the kinematic arrangement diagram of an existing gearbox and compare the results with the solution provided by the program.

If option 'EX' entered the program types out proportional elasticity and inertia, percentage error range on output speeds and maximum gear ratio of all the executed options for comparison purposes. Program turns back to the beginning. If again 'EX' is typed, the execution of program terminates.

4.3 ILLUSTRATIVE EXAMPLES

To show the use and integrity of the computer program 6 examples are presented. The examples are taken from previous literature and from the machine tools available in the market.

EXAMPLES 1 and 2

As it is explained in the preceeding chapters, kinematical arrangement diagrams are traditionally prepared with graphical methods, Germar prepared kinematical arrangement diagrams for 18 speed 3 shafts and 12 speed 3 shafts gear boxes given in reference [1]. His results are re-drawn in Figures (4.1) and (4.2) respectively.

The computer program run with suitable input data to obtain the similar solutions for the two gear boxes above. The input data for the two examples are given in Tables (4.1) and (4.3). The results of the computer program can be seen in Tables (4.2) and (4.4). With the use of the information given under the heading "Power of Progression Ratio" (POW.of Prog. Rt.) the similar kinematical arrangement diagrams can be drawn. For comparison purpose, the option numbers of output results are written on top of each diagram (Figures 4.1 and 4.2)

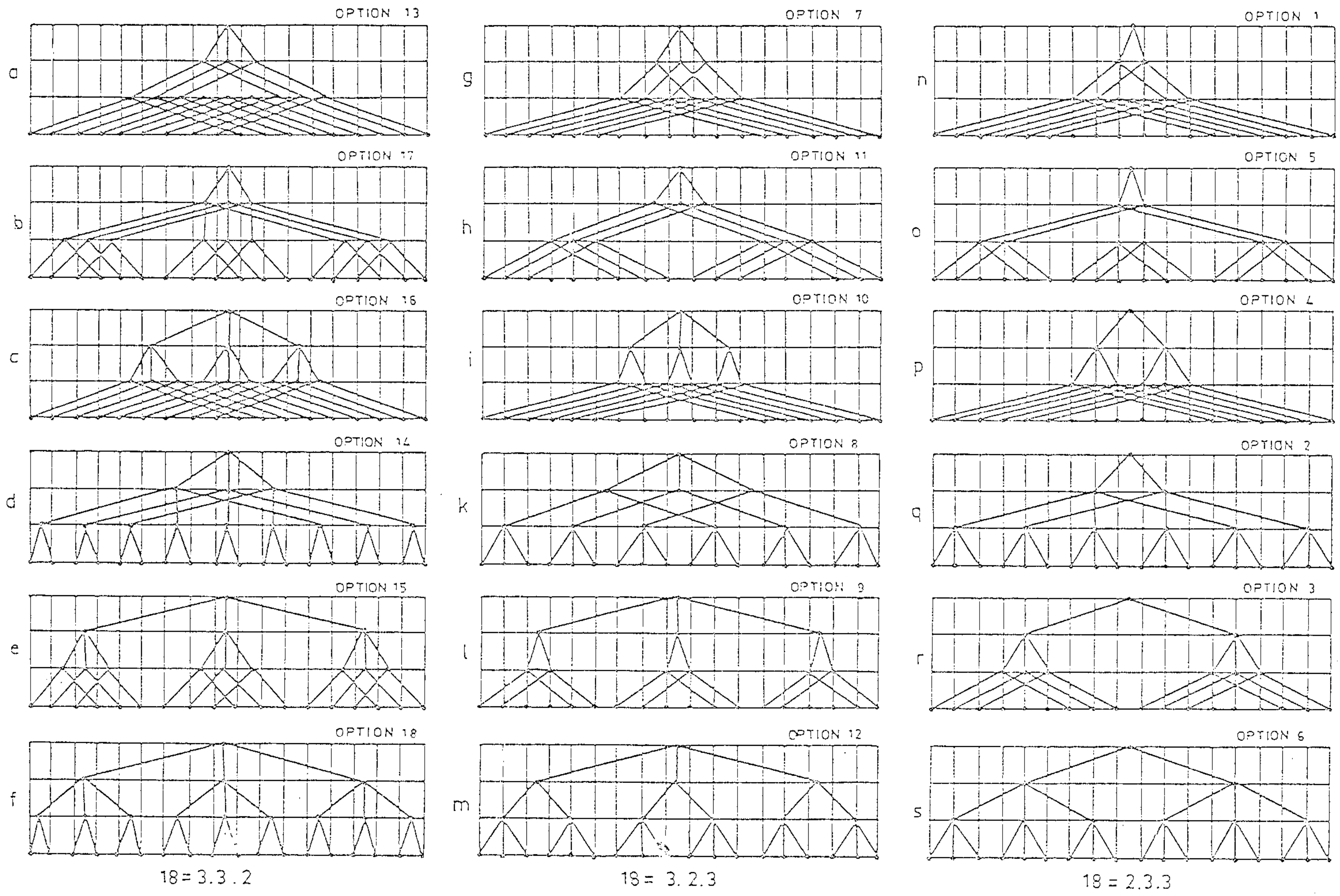


Figure 4.1 All Kinematical Arrangement Diagrams of 18 Speed 4 Shafts Gearbox

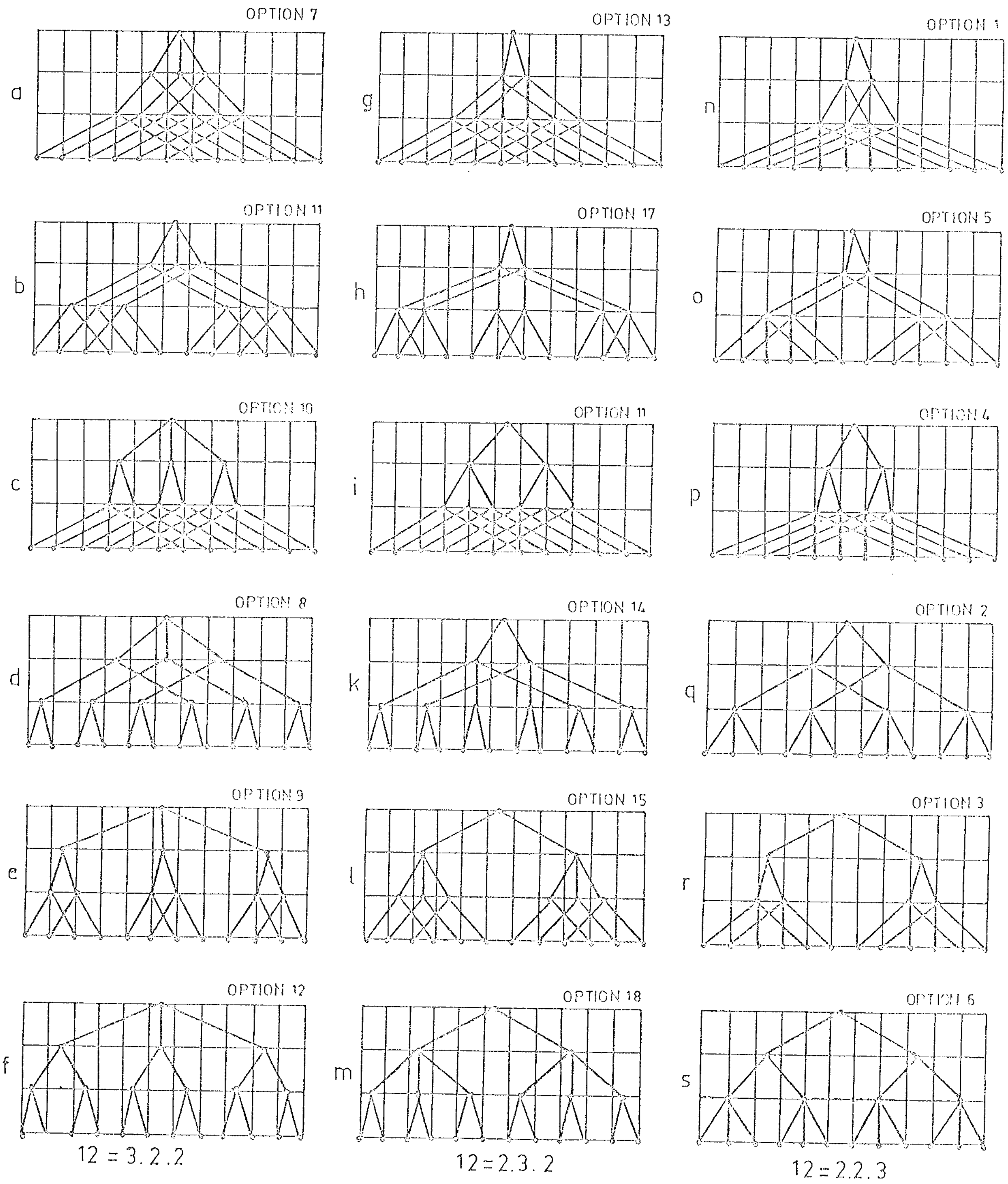


Figure 4.2 All Kinematical Arrangement Diagrams of 12 Speed 4 Shafts Gearbox

Table 4.1 Computer Input Data for Example 1

```
INPUT VALUES

  NUMBER OF SPEEDS              =    18
  NUMBER OF SHAFTS              =     4
  SHAFTS WITH 1 TRANS.          =     0
  INPUT SPEED                   = 1400.00
  MINIMUM SPEED                 =   84.00
  MAXIMUM SPEED                 =  576.74
  PROGRESSION RATIO             =     1.1200

LIMITATIONS

  MINIMUM TEETH NUMBER          =    18
  TOTAL TEETH NUMBER            =   120
  % ERROR ON GEAR RATIO         = \+ 1.00 %
  % ERROR ON OUTPUT SPEED       = \+ 2.00 %
  MINIMUM TRANS. RATIO          =     0.25
  MAXIMUM TRANS. RATIO          =     2.40
```

Table 4.2 Computer Output Results for Example 1

```
                 E X A M P L E   1

           ALL KINEMATICAL ARRANGEMENTS


   FOR  18 SPEED    4 SHAFTS    GEARBOX
```

Table 4.2 cont'd.

OPTION 1 ****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		1
2	3	0
		2
		4
3	3	0
		6
		12

*** OPTION 2 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		3
2	3	0
		6
		12
3	3	0
		1
		2

** OPTION 3 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		9
2	3	0
		1
		2
3	3	0
		3
		6

* OPTION 4 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		3
2	3	0
		1
		2
3	3	0
		6
		12

** OPTION 5 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		1
2	3	0
		6
		12
3	3	0
		2
		4

*** OPTION 6 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		9
2	3	0
		3
		6
3	3	0
		1
		2

*** OPTION 7 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		1
		2
2	2	0
		3
3	3	0
		6
		12

*** OPTION 8 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		3
		6
2	2	0
		9
3	3	0
		1
		2

*** OPTION 9 *

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		6
		12
2	2	0
		1
3	3	0
		2
		4

Table 4.2 cont'd.

** OPTION 10 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		2
		4
2	2	0
		1
3	3	0
		6
		12

** OPTION 11 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		1
		2
2	2	0
		9
3	3	0
		3
		6

** OPTION 12 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		6
		12
2	2	0
		3
3	3	0
		1
		2

** OPTION 13 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		1
		2
2	3	0
		3
		6
3	2	0
		9

** OPTION 14 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		2
		4
2	3	0
		6
		12
3	2	0
		1

** OPTION 15 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		6
		12
2	3	0
		1
		2
3	2	0
		3

** OPTION 16 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		3
		6
2	3	0
		1
		2
3	2	0
		9

** OPTION 17 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		1
		2
2	3	0
		6
		12
3	2	0
		3

** OPTION 18 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		6
		12
2	3	0
		2
		4
3	2	0
		1

Table 4.3 Computer Input Data for Example 2

```
INPUT VALUES

  NUMBER OF SPEEDS              =     12
  NUMBER OF SHAFTS              =      4
  SHAFTS WITH 3 TRANS.          =      3
  INPUT SPEED                   =  1400.00
  MINIMUM SPEED                 =    53.00
  MAXIMUM SPEED                 =   673.52
  PROGRESSION RATIO             =      1.2600

LIMITATIONS

  MINIMUM TEETH NUMBER          =     18
  TOTAL TEETH NUMBER            =    120
  % ERROR ON GEAR RATIO         = \+ 1.00 %
  % ERROR ON OUTPUT SPEED       = \+ 2.00 %
  MINIMUM TRANS. RATIO          =      0.25
  MAXIMUM TRANS. RATIO          =      2.40
```

Table 4.4 Computer Output Results for Example 2

```
          E X A M P L E   2

    ALL KINEMATICAL ARRANGEMENTS


  FOR  12 SPEED  4 SHAFTS  GEARBOX
```

Table 4.4 cont'd.

** OPTION 1 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	01
2	2	02
3	3	048

OPTION 2 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	03
2	2	06
3	3	012

OPTION 3 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	06
2	2	01
3	3	024

OPTION 4 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	02
2	2	01
3	3	048

OPTION 5 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	01
2	2	04
3	3	024

OPTION 6 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	06
2	2	03
3	3	012

OPTION 7 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	012
2	2	03
3	2	06

OPTION 8 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	024
2	2	06
3	2	01

OPTION 9 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	048
2	2	01
3	2	02

Table 4.4 cont'd.

OPTION 10 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		2
		4
2	2	0
		1
3	2	0
		6

OPTION 11 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		1
		2
2	2	0
		6
3	2	0
		3

OPTION 12 ****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	3	0
		4
		8
2	2	0
		2
3	2	0
		1

OPTION 13 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		1
2	3	0
		2
		4
3	2	0
		6

OPTION 14 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		2
2	3	0
		4
		8
3	2	0
		1

OPTION 15 ****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		6
2	3	0
		1
		2
3	2	0
		3

OPTION 16 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		3
2	3	0
		1
		2
3	2	0
		6

OPTION 17 **

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		1
2	3	0
		4
		8
3	2	0
		2

OPTION 18 ***

GRUP NUMB	SPED NUMB	POW.OF PRG.RT
1	2	0
		6
2	3	0
		2
		4
3	2	0
		1

EXAMPLES 3 and 4

Koenigsberger [1] analysed the kinematical arrangement diagrams of 18 speed 5 shafts and 12 speed 5 shafts gear boxes. He gave the gear teeth numbers and calculated percentage error on each output speed. Speed charts and gear teeth numbers are given in Figures (4.3) and (4.4) respectively for the two gear boxes.

The relevant input data and limitations are presented in Tables (4.5) and (4.7). The results of computer program and the analysis of above examples are given in Tables (4.6) and (4.8).

Comparison of the results show that prepared computer program gives minimum teeth numbers within specified limits. Percentage error ranges on output speeds are smaller than the actual arrangements given by [1].

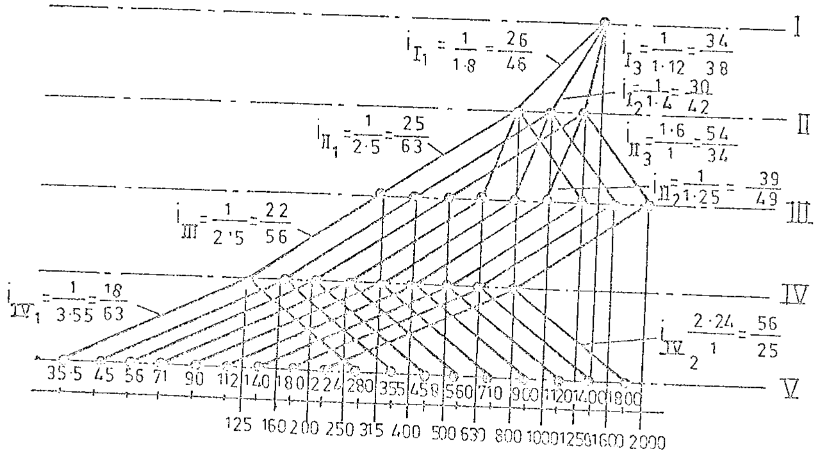


Figure 4.3 Speed Chart of 18 Speed 5 Shafts Gearbox

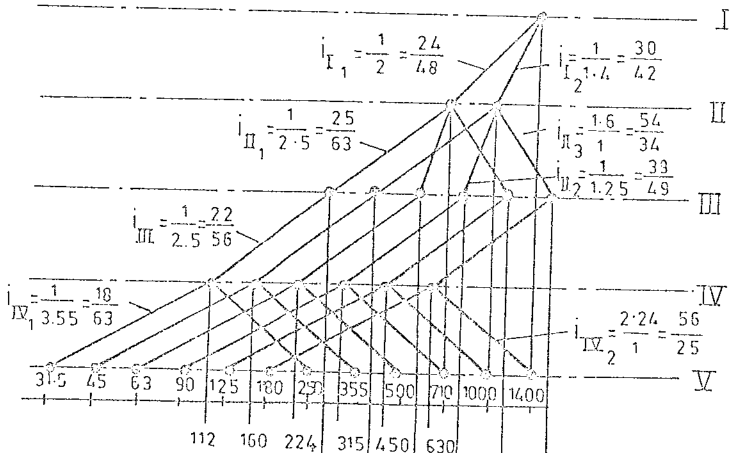


Figure 4.4 Speed Chart of 12 Speed 5 Shafts Gearbox

Table 4.5 Computer Input Data for Example 3

```
INPUT VALUES

  NUMBER OF SPEEDS            =    18
  NUMBER OF SHAFTS            =     5
  SHAFTS WITH 1 TRANS.        =     1
  INPUT SPEED                 = 1400.00
  MINIMUM SPEED               =   35.50
  MAXIMUM SPEED               = 1805.21
  PROGRESSION RATIO           =     1.2600

LIMITATIONS

  MINIMUM TEETH NUMBER        =   18
  TOTAL TEETH NUMBER          =  120
  % ERROR ON GEAR RATIO       = \+ 1.00 %
  % ERROR ON OUTPUT SPEED=      \+ 2.00 %
  MINIMUM TRANS. RATIO        =     0.25
  MAXIMUM TRANS. RATIO        =     2.40
```

Table 4.7 Computer Input Data for Example 4

```
INPUT VALUES

  NUMBER OF SPEEDS            =    12
  NUMBER OF SHAFTS            =     5
  SHAFTS WITH 1 TRANS.        =     1
  INPUT SPEED                 = 1400.00
  MINIMUM SPEED               =   31.50
  MAXIMUM SPEED               = 1423.15
  PROGRESSION RATIO           =     1.4140

LIMITATIONS

  MINIMUM TEETH NUMBER        =   18
  TOTAL TEETH NUMBER          =  120
  % ERROR ON GEAR RATIO       = \+ 1.00 %
  % ERROR ON OUTPUT SPEED=      \+ 2.00 %
  MINIMUM TRANS. RATIO        =     0.25
  MAXIMUM TRANS. RATIO        =     2.40
```

Table 4.6 Computer Output Results for Example 3

E X A M P L E   3

TO COMPARE THE RESULTS WITH

THE EXAMPLE GIVEN BY KONIGSBERGER

FOR 18 SPEED 4 SHAFTS GEARBOX

************************* OPTION 13 *******************************

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	3	0	0.570000	21 / 37	0.567568	\0.426752
		1	0.718200	23 / 32	0.718750	0.076568
		2	0.904932	29 / 32	0.906250	0.145664
2	3	0	0.400000	18 / 45	0.400000	\0.000009
		3	0.800150	28 / 35	0.800000	\0.018773
		6	1.600600	40 / 25	1.599999	\0.037501
3	1	0	0.400000	18 / 45	0.400000	\0.000009
4	2	0	0.280000	19 / 68	0.279412	\0.210095
		9	2.241261	58 / 26	2.230769	\0.466093

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	35.75	35.52	\0.64	0.36908803E\08	0.51523552E+08
2	45.05	44.99	\0.13	0.41966217E\08	0.58933920E+08
3	56.76	56.72	\0.06	0.43186912E\08	0.10679618E+09
4	71.51	71.05	\0.66	0.36224217E\08	0.70311952E+08
5	90.11	89.97	\0.15	0.40868393E\08	0.89064672E+08
6	113.54	113.44	\0.08	0.46441535E\08	0.15469781E+09
7	143.06	142.09	\0.67	0.68426527E\08	0.14018398E+09
8	180.25	179.94	\0.17	0.92831698E\08	0.20111776E+09
9	227.11	226.88	\0.10	0.12905247E\07	0.33283866E+09
10	286.16	283.61	\0.89	0.37184746E\08	0.57526560E+08
11	360.57	359.15	\0.39	0.42408743E\08	0.68560896E+08
12	454.31	452.35	\0.32	0.48990421E\08	0.12210102E+09
13	572.44	567.22	\0.91	0.37327972E\08	0.94324016E+08
14	721.27	718.31	\0.41	0.42638497E\08	0.12757248E+09
15	908.50	905.39	\0.34	0.49755640E\08	0.21591728E+09
16	1145.07	1134.44	\0.93	0.73041626E\08	0.23623211E+09
17	1442.81	1436.61	\0.43	0.99712150E\08	0.35514906E+09
18	1817.94	1811.38	\0.36	0.14039892E\07	0.57771648E+09

TOTAL ELASTICITY CONSTANT = 0.11120738E\06

TOTAL INERTIA CONSTANT = 0.30605473E+10

SUM OF LARGEST GEAR RATIOS = 5.137019

MAXIMUM GEAR RATIO OF OPTION = 2.230769

ERROR RANGE OF THIS OPTION = 0.932369

****

Table 4.6 cont'd.

************************ OPTION 13 ******************************

***** TEETH NUMBERS CHANGED *****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	3	0	0.570000	26 / 46	0.565217	0.839155
		1	0.718200	30 / 42	0.714286	0.545031
		2	0.904932	34 / 38	0.894737	1.126606
2	3	0	0.400000	25 / 63	0.396825	0.793651
		3	0.800150	39 / 49	0.795918	0.528884
		6	1.600600	54 / 34	1.588235	0.772544
3	1	0	0.400000	22 / 56	0.392857	1.785710
4	2	0	0.280000	18 / 63	0.285714	\2.040820
		9	2.241261	56 / 25	2.240000	0.056252

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	35.75	35.25	1.38	0.18891908E\08	0.16782387E+09
2	45.05	44.54	1.08	0.17066173E\08	0.22822760E+09
3	56.76	55.79	1.67	0.23572659E\08	0.31587610E+09
4	71.51	70.69	1.11	0.19927039E\08	0.23062112E+09
5	90.11	89.34	0.82	0.20719384E\08	0.32951610E+09
6	113.54	111.91	1.40	0.26166695E\08	0.47323827E+09
7	143.06	141.07	1.36	0.38608086E\08	0.42350848E+09
8	180.25	178.27	1.06	0.59553446E\08	0.63656320E+09
9	227.11	223.31	1.64	0.72778921E\08	0.95659034E+09
10	286.16	276.33	3.47	0.19176261E\08	0.17271867E+09
11	360.57	349.21	3.18	0.19520299E\08	0.23604470E+09
12	454.31	437.43	3.76	0.24285218E\08	0.32814157E+09
13	572.44	554.24	3.21	0.21071020E\08	0.25031227E+09
14	721.27	700.41	2.92	0.22546278E\08	0.35976314E+09
15	908.80	877.35	3.50	0.27033242E\08	0.52259176E+09
16	1145.09	1105.96	3.45	0.43163162E\08	0.50191718E+09
17	1442.81	1397.65	3.16	0.57828018E\08	0.76178406E+09
18	1817.94	1750.74	3.74	0.84393292E\08	0.11530729E+10

TOTAL ELASTICITY CONSTANT = 0.61150047E\07

TOTAL INERTIA CONSTANT = 0.80474849E+10

SUM OF LARGEST GEAR RATIOS = 5.137019

MAXIMUM GEAR RATIO OF OPTION = 2.230769

ERROR RANGE OF THIS OPTION = 3.762219

****

Table 4.8 Computer Output Results for Example 4

E X A M P L E 4

TO COMPARE THE RESULTS WITH

THE EXAMPLE GIVEN BY KOENIGSBERGER

FOR 12 SPEED 5 SHAFTS GEARBOX

************************ OPTION 13 ******************************

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	2	0	0.500000	18 / 36	0.500000	\0.000012
		1	0.707000	22 / 31	0.709677	0.378733
2	3	0	0.400000	18 / 45	0.400000	\0.000009
		2	0.799758	28 / 35	0.800000	0.030294
		4	1.599930	40 / 25	1.599999	0.060631
3	1	0	0.400000	18 / 45	0.400000	\0.000009
4	2	0	0.280000	19 / 68	0.279412	\0.210075
		6	2.237965	58 / 26	2.230769	\0.321515

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	31.36	31.29	\0.21	0.45374513E\08	0.35444096E+08
2	44.34	44.42	0.17	0.43942308E\08	0.52724672E+08
3	62.70	62.59	\0.18	0.44843205E\08	0.50025312E+08
4	88.66	88.83	0.20	0.42872053E\08	0.82099568E+08
5	125.36	125.18	\0.15	0.69789987E\08	0.10425142E+09
6	177.26	177.67	0.23	0.93531796E\08	0.19134170E+09
7	250.65	249.85	\0.32	0.45589671E\08	0.40102696E+08
8	354.42	354.62	0.06	0.44373749E\08	0.62110128E+08
9	501.15	499.69	\0.29	0.45699035E\08	0.68660528E+08
10	708.63	709.24	0.09	0.44597748E\08	0.11964147E+09
11	1002.09	999.38	\0.26	0.73416492E\08	0.17879218E+09
12	1416.83	1418.48	0.12	0.10043461E\07	0.34150886E+09

TOTAL ELASTICITY CONSTANT = 0.69466466E\07

TOTAL INERTIA CONSTANT = 0.13267018E+10

SUM OF LARGEST GEAR RATIOS = 4.940445

MAXIMUM GEAR RATIO OF OPTION = 2.230769

ERROR RANGE OF THIS OPTION = 0.550805

****

Table 4.8 cont'd.

************************ OPTION 13 ******************************

***** TEETH NUMBERS CHANGED *****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	2	0	0.500000	24 / 48	0.500000	0.000000
		1	0.707000	30 / 42	0.714286	\1.030537
2	3	0	0.400000	25 / 63	0.396825	0.793651
		2	0.799758	39 / 49	0.795918	0.480067
		4	1.599030	54 / 34	1.588235	0.675134
3	1	0	0.400000	22 / 56	0.392857	1.785710
4	2	0	0.280000	18 / 63	0.285714	\2.040829
		6	2.237965	56 / 25	2.240000	\0.090937

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	31.36	31.18	0.54	0.20487911E\08	0.14416653E+09
2	44.34	44.54	\0.49	0.19066173E\08	0.22822760E+09
3	62.70	62.54	0.22	0.21297992E\08	0.19330824E+09
4	88.66	89.34	\0.81	0.20719384E\08	0.32851610E+09
5	125.36	124.79	0.42	0.35916736E\08	0.34425114E+09
6	177.26	179.27	\0.61	0.50553446E\08	0.63656320E+09
7	250.65	244.44	2.49	0.20710429E\08	0.14799690E+09
8	354.42	349.21	1.46	0.19520299E\08	0.23604470E+09
9	501.15	490.29	2.17	0.22193170E\08	0.20871747E+09
10	708.63	700.41	1.14	0.22546278E\08	0.35996314E+09
11	1002.00	978.35	2.37	0.39481236E\08	0.40560947E+09
12	1416.83	1397.65	1.34	0.57828018E\08	0.76178406E+09

TOTAL ELASTICITY CONSTANT = 0.35032095E\07

TOTAL INERTIA CONSTANT = 0.39951478E+10

SUM OF LARGEST GEAR RATIOS = 4.940445

MAXIMUM GEAR RATIO OF OPTION = 2.230769

ERROR RANGE OF THIS OPTION = 3.294003

****

EXAMPLE 5

The gear box used on the main drive of Ajax Lathe is analysed with this example and the computer results are obtained. Kinematic arrangement diagram and speed chart of the gear box are given in Fig. (3.15) and (3.16).

The gear box is designed to provide 24 speeds. 12 of the speeds are obtained with normal kinematical arrangement and the other 12 speeds are obtained with modified arrangements and not included in Fig. (3.15) and (3.16).

The input data for the example is given in Table (4.9), and results are presented in Table (4.10) with the comparison of the actual arrangement and computer results, it can be seen that gear teeth numbers are minimum, percentage error on output speeds are minimum for the computer results.

Table 4.9 Computer Input Data for Example 5

```
INPUT VALUES

  NUMBER OF SPEEDS              =      12
  NUMBER OF SHAFTS              =       6
  SHAFTS WITH 1 TRANS.          =       2
  INPUT SPEED                   =  1040.00
  MINIMUM SPEED                 =    12.50
  MAXIMUM SPEED                 =   188.99
  PROGRESSION RATIO             =     1.2800

LIMITATIONS

  MINIMUM TEETH NUMBER          =     18
  TOTAL TEETH NUMBER            =    120
  % ERROR ON GEAR RATIO         = \+ 1.00 %
  % ERROR ON OUTPUT SPEED       = \+ 2.00 %
  MINIMUM TRANS. RATIO          =     0.25
  MAXIMUM TRANS. RATIO          =     2.00
```

Table 4.10 Computer Output Results for Example 5

E X A M P L E 5

TO COMPARE THE RESULTS

WITH THE GEARBOX ARRANGEMENTS

OF A J A X L A T H E

************************ OPTION 89 ********************************

GRUP NUMB	SPED NUMB	PUW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	2	0	0.827000	19 / 23	0.826087	\0.110414
		3	1.734343	26 / 15	1.733333	\0.058192
2	3	0	0.517000	18 / 35	0.514286	\0.525022
		1	0.661760	20 / 30	0.666667	0.741479
		2	0.847052	23 / 27	0.851852	0.566614
3	1	0	0.250000	18 / 72	0.250000	\0.000024
4	2	0	0.250000	18 / 72	0.250000	\0.000024
		6	1.099509	46 / 42	1.095238	\0.388459
5	1	0	0.440000	18 / 41	0.439024	\0.221742

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	12.23	12.12	\0.96	0.71919359E\08	0.52680208E+08
2	15.65	15.72	0.41	0.72155899E\08	0.70202688E+08
3	20.03	20.08	0.23	0.76571993E\08	0.10228339E+09
4	25.64	25.44	\0.81	0.33582584E\07	0.47169869E+09
5	32.82	32.98	0.46	0.33582584E\07	0.47169869E+09
6	42.02	42.14	0.29	0.33423863E\07	0.42447078E+09
7	53.73	53.11	\1.25	0.72050739E\08	0.56590076E+08
8	68.84	68.85	0.02	0.72375736E\08	0.76772816E+08
9	88.11	87.98	\0.15	0.76932523E\08	0.11301058E+09
10	112.78	111.44	\1.19	0.33582584E\07	0.47169869E+09
11	144.36	144.46	0.07	0.33582584E\07	0.47169869E+09
12	184.79	184.59	\0.10	0.33582584E\07	0.47169869E+09

TOTAL ELASTICITY CONSTANT = 0.23707219E\06

TOTAL INERTIA CONSTANT = 0.23924933E+10

SUM OF LARGEST GEAR RATIOS = 4.370421

MAXIMUM GEAR RATIO OF OPTION = 1.733333

ERROR RANGE OF THIS OPTION = 1.707157

****

Table 4.10 cont'd.

************************ OPTION 89 ******************************

***** TEETH NUMBERS CHANGED *****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	2	0	0.827000	43 / 52	0.826923	0.009305
		3	1.734343	58 / 35	1.657143	4.451250
2	3	0	0.517000	30 / 58	0.517241	\0.046681
		1	0.661760	35 / 53	0.660377	0.208917
		2	0.847052	40 / 48	0.833333	1.619618
3	1	0	0.250000	15 / 60	0.250000	0.000000
4	2	0	0.250000	14 / 56	0.250000	0.000000
		6	1.099502	35 / 35	1.000000	9.050327
5	1	0	0.440000	22 / 50	0.440000	0.000000

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	12.23	12.23	\0.04	0.20972597E\08	0.67497600E+09
2	15.65	15.62	0.22	0.26099192E\08	0.71437619E+09
3	20.03	19.71	1.63	0.36016891E\08	0.76552525E+09
4	25.64	24.51	4.40	0.13951599E\07	0.20609961E+10
5	32.82	31.30	4.66	0.13951599E\07	0.20609961E+10
6	42.02	39.30	6.07	0.14391745E\07	0.20403656E+10
7	53.79	48.93	9.01	0.20559361E\08	0.67695514E+09
8	68.84	62.47	9.27	0.25410929E\08	0.71760295E+09
9	88.11	78.83	10.68	0.34929891E\08	0.77066266E+09
10	112.78	98.06	13.45	0.13951599E\07	0.20609961E+10
11	144.36	125.19	13.71	0.13951599E\07	0.20609961E+10
12	184.79	157.98	15.12	0.13951599E\07	0.20609961E+10

TOTAL ELASTICITY CONSTANT = 0.81612113E\07

TOTAL INERTIA CONSTANT = 0.15465701E+11

SUM OF LARGEST GEAR RATIOS = 4.370421

MAXIMUM GEAR RATIO OF OPTION = 1.733333

ERROR RANGE OF THIS OPTION = 15.158571

****

EXAMPLE 6

This run is used to compare and analyze the computer results with the main drive gear box of TOS Universal Milling Machine. This gear box provides 12 speeds with 5 shafts. Kinematic arrangement diagram and speed chart of the gear box are given in Figures (4.5) and (4.6). The input data and limitations are introduced in Table (4.11) and the results are given in Table (4.12).

With a detailed study on the results, some conclusions can be reached.

Table 4.11 Computer Input Data for Example 6

```
INPUT VALUES

  NUMBER OF SPEEDS            =    12
  NUMBER OF SHAFTS            =     5
  SHAFTS WITH 1 TRANS.        =     1
  INPUT SPEED                 = 1400.00
  MINIMUM SPEED               =   42.00
  MAXIMUM SPEED               = 1989.02
  PROGRESSION RATIO           =     1.4200

LIMITATIONS

  MINIMUM TEETH NUMBER        =    19
  TOTAL TEETH NUMBER          =   120
  % ERROR ON GEAR RATIO       = \+ 1.00 %
  % ERROR ON OUTPUT SPEED     = \+ 2.00 %
  MINIMUM TRANS. RATIO        =     0.25
  MAXIMUM TRANS. RATIO        =     2.20
```

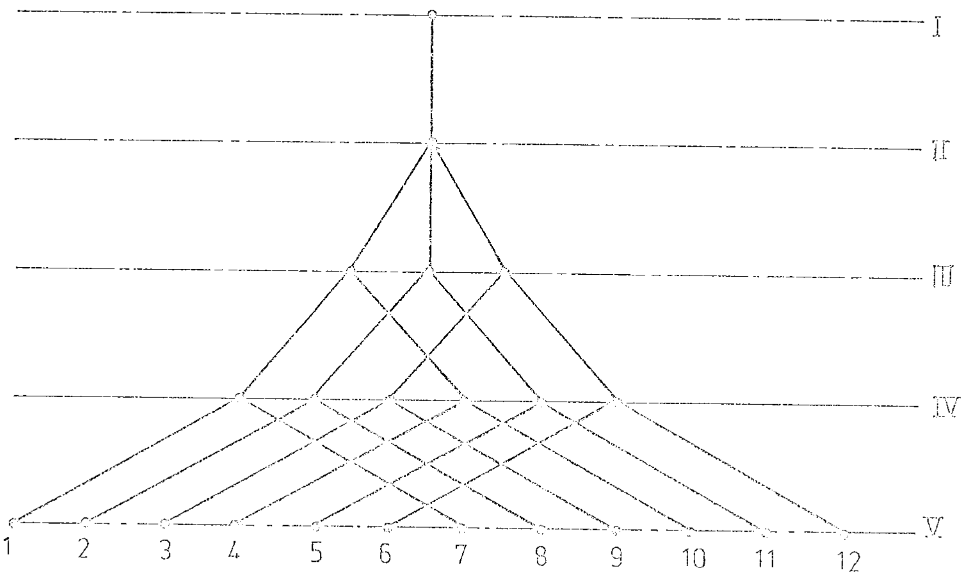


Figure 4.5 Kinematic Arrangement Diagram of the Gearbox used for Main Drive of TOS Milling Machine

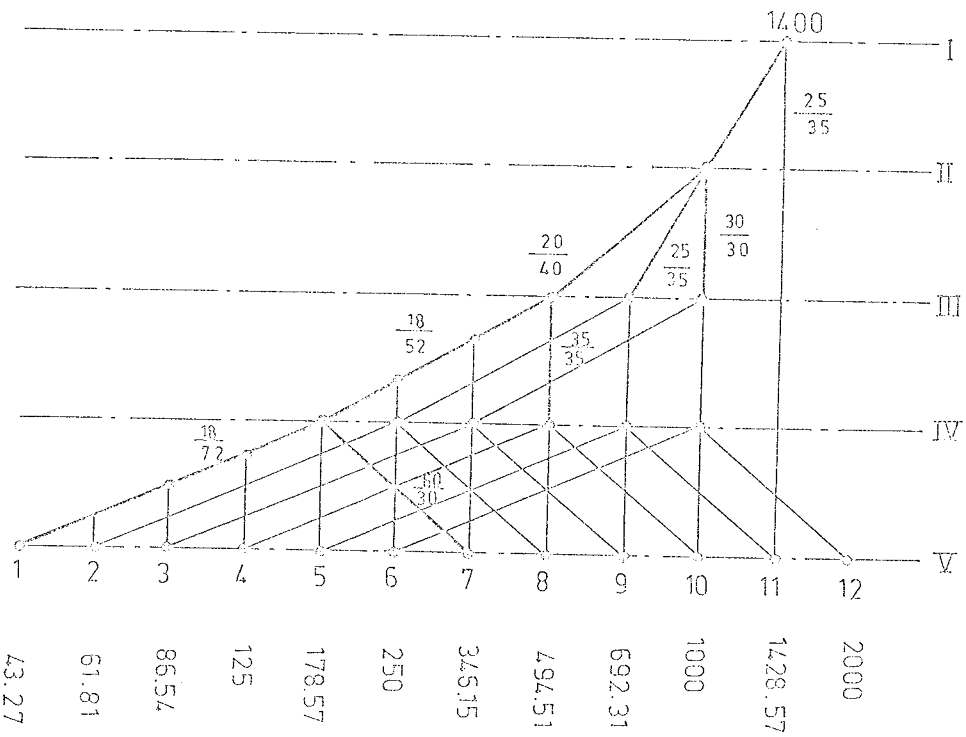


Figure 4.6 Speed Chart of the Gearbox used for Main Drive of TOS Milling Machine

Table 4.12 Computer Output Results for Example 6

E X A M P L E 6

TO COMPARE THE RESULTS WITH

THE GEARBOX ARRANGEMENTS OF

T O S UNIVERSAL MILLING MACHINE

************************ OPTION 43 ******************************

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBERS	FOUND RATIO	PERCENTAGE ERROR
1	1	0	0.714000	22 / 31	0.709677	\0.605410
2	3	0	0.500000	18 / 36	0.500000	\0.000012
		1	0.710000	22 / 31	0.709677	\0.045439
		2	1.008200	26 / 26	1.000000	\0.813300
3	2	0	0.346000	19 / 55	0.345455	\0.157668
		3	0.990697	36 / 36	1.000000	0.939003
4	2	0	0.250000	18 / 72	0.250000	\0.000024
		6	2.049603	59 / 29	2.034482	\0.737692

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	43.23	42.90	\0.76	0.45432742E\08	0.50947872E+08
2	61.39	60.89	\0.81	0.43909832E\08	0.65154192E+08
3	87.17	85.81	\1.58	0.52859000E\08	0.91756512E+09
4	123.79	124.19	0.33	0.46734634E\08	0.77978704E+08
5	175.78	176.27	0.29	0.46431537E\08	0.11779651E+09
6	249.61	248.39	\0.48	0.58062094E\08	0.19627970E+09
7	354.44	349.14	\1.50	0.45507029E\08	0.63287312E+08
8	503.30	495.56	\1.55	0.43958590E\08	0.90012564E+08
9	714.59	698.29	\2.31	0.53152149E\08	0.14111435E+09
10	1014.86	1019.68	\0.40	0.47357105E\08	0.18047726E+09
11	1441.10	1434.91	\0.45	0.47685660E\08	0.32409946E+09
12	2046.37	2021.36	\1.22	0.60552594E\08	0.60987341E+09

TOTAL ELASTICITY CONSTANT = 0.59153844E\07

TOTAL INERTIA CONSTANT = 0.20098772E+10

SUM OF LARGEST GEAR RATIOS = 4.748482

MAXIMUM GEAR RATIO OF OPTION = 2.034482

ERROR RANGE OF THIS OPTION = 2.647625

****

Table 4.12 cont'd.

************************ OPTION 43 *********************************

***** TEETH NUMBERS CHANGED *****

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	1	0	0.714000	25 / 35	0.714286	\0.040012
2	1	0	0.500000	20 / 40	0.500000	0.000000
		1	0.710000	25 / 35	0.714286	\0.603610
		2	1.003200	30 / 30	1.000000	0.313300
3	2	0	0.346000	18 / 52	0.346154	\0.044445
		3	0.990697	35 / 35	1.000000	\0.939003
4	2	0	0.250000	18 / 72	0.250000	0.000000
		6	2.049603	60 / 30	2.000000	2.420103

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	43.23	43.27	\0.08	0.34916858E\08	0.78621232E+08
2	61.39	61.31	\0.69	0.34946639E\08	0.94579440E+08
3	87.17	86.54	0.73	0.44529647E\08	0.12234400E+09
4	123.79	125.00	\0.98	0.35744765E\08	0.10481426E+09
5	175.78	178.57	\1.58	0.36636243E\08	0.14813454E+09
6	249.61	250.00	\0.17	0.47840274E\08	0.22711602E+09
7	354.44	346.15	2.34	0.34721477E\08	0.92505648E+08
8	503.30	494.51	1.73	0.34956069E\08	0.12301493E+09
9	714.69	692.31	3.15	0.44547122E\08	0.17788158E+09
10	1014.86	1000.00	1.44	0.35783310E\08	0.22068912E+09
11	1441.19	1428.57	0.34	0.36714927E\08	0.38461338E+09
12	2046.37	2000.00	2.25	0.47094497E\08	0.69061478E+09

TOTAL ELASTICITY CONSTANT = 0.46953059E\07

TOTAL INERTIA CONSTANT = 0.24650278E+10

SUM OF LARGEST GEAR RATIOS = 4.748482

MAXIMUM GEAR RATIO OF OPTION = 2.034482

ERROR RANGE OF THIS OPTION = 4.731570

****

CHAPTER 5

5.1. DISCUSSION AND CONCLUSION

In the previous chapters, the theory of kinematical arrangement diagrams are discussed. The construction of kinematical arrangement diagrams in the previous works were done by graphical methods. As it is mentioned, there are many possible solutions and with graphical methods all of the solutions can not be obtained for large number of shafts.

Investigations on the examples given in the preceeding chapter proved that all of the possible and different solutions can be obtained by use of computer programs. With the use of these results, kinematic arrangement diagrams can be drawn without any difficulty. And other parameters related to kinematic arrangement diagrams are calculated. Recommended options are presented according to preset limitations.

In construction of kinematical arrangement diagrams, determination of transmission numbers between the groups and order of arrangement of groups is critical. When these values are established transmission ratios of the groups are determined by taking the minimum transmission ratios of each group into account.

With the examples given, it is shown that the best kinematical arrangement is obtained when the order of arrangement is parallel to the order of groups in the gear box. It is option 13, for the example 1 with speed distribution of 3x3x2 and with the order of arrangement 1x2x3. For the example 2, it is option 7 with speed distribution of 3x2x2 and arrangement of 1x2x3, which contains minimum values for the maximum gear ratio and sum of maximum gear ratios in the groups. These results prove the recommendations given in Section 3.7 about speed distribution and order of arrangements. This fact is also stated by reference [1].

Gear teeth numbers when calculated with Gray Method for minimum transmission ratios, quite satisfactory results are obtained. One thing to be mentioned here is that in some cases, computer programs may give unexpected results in determining the gear teeth numbers. For example for a transmission ratio of 0.4, the gear teeth numbers are found as 22/56, giving a ratio of 0.3928571. But the method gives 18/45= 0.4 with no error. This is due to the truncation error in conversions of real numbers to its binary equivalent form. To overcome this problem, the value of transmission ratio has to be increased by a very small quantity. (i.e. 1×10^{-7}).

For fixed center distance, gear teeth numbers are calculated with the use of the method explained in Section 3.8.2. As it can be seen on the output results, some of gear teeth numbers are given as 0/0. This means that integer gear teeth numbers are not found for specified center distance, gear ratio and percentage error.

The iteration to obtain all possible solutions are carried out one-by-one. When one iteration is completed, the user has the opportunity of entering data of a kinematic arrangement diagram of a known or used gear box to analyse and compare with the results of computer program.

At the end of the iterations the data; total proportional elasticity, total proportional inertia, error range of output speeds, maximum transmission ratio in the arrangement and sum of maximum transmission ratios of each group are printed out which are helpful in the selection of suitable kinematic arrangement diagrams.

A fully automated process is developed for the determination of all possible kinematical arrangement diagrams and gear teeth numbers.

Informative, desicive and error messages are inserted into the program to aid the user in the design process and input values are controlled against unlogical values.

The inputs and outputs are put in an organized form to follow the results easily.

The recommended options are found which do not contain gear ratios out of limits. Other properties mentioned above and related to kinematical arrangement diagrams are calculated and it is left to the user to select one kinematic arrangement diagram between recommended options according to the required constraints.

5.2 SUGGESTIONS FOR FUTURE WORK

The work presented in this thesis can be extended to the following fields to have a complete design package for gear boxes.

A- To increase the range ratio of output speeds following modifications on the kinematic arrangement diagrams can be done

- use of two geometric progression ratio for the same kinematic arrangement diagram
- use of broken geometrical series
- use of overlapping speed steps

B- To complete the design of gear box the other design stages, bending strength design surface fatigue, strength design, shaft design, bearing selection, lubricant selection and torsional analysis, all must be integrated with the prepared computer program.

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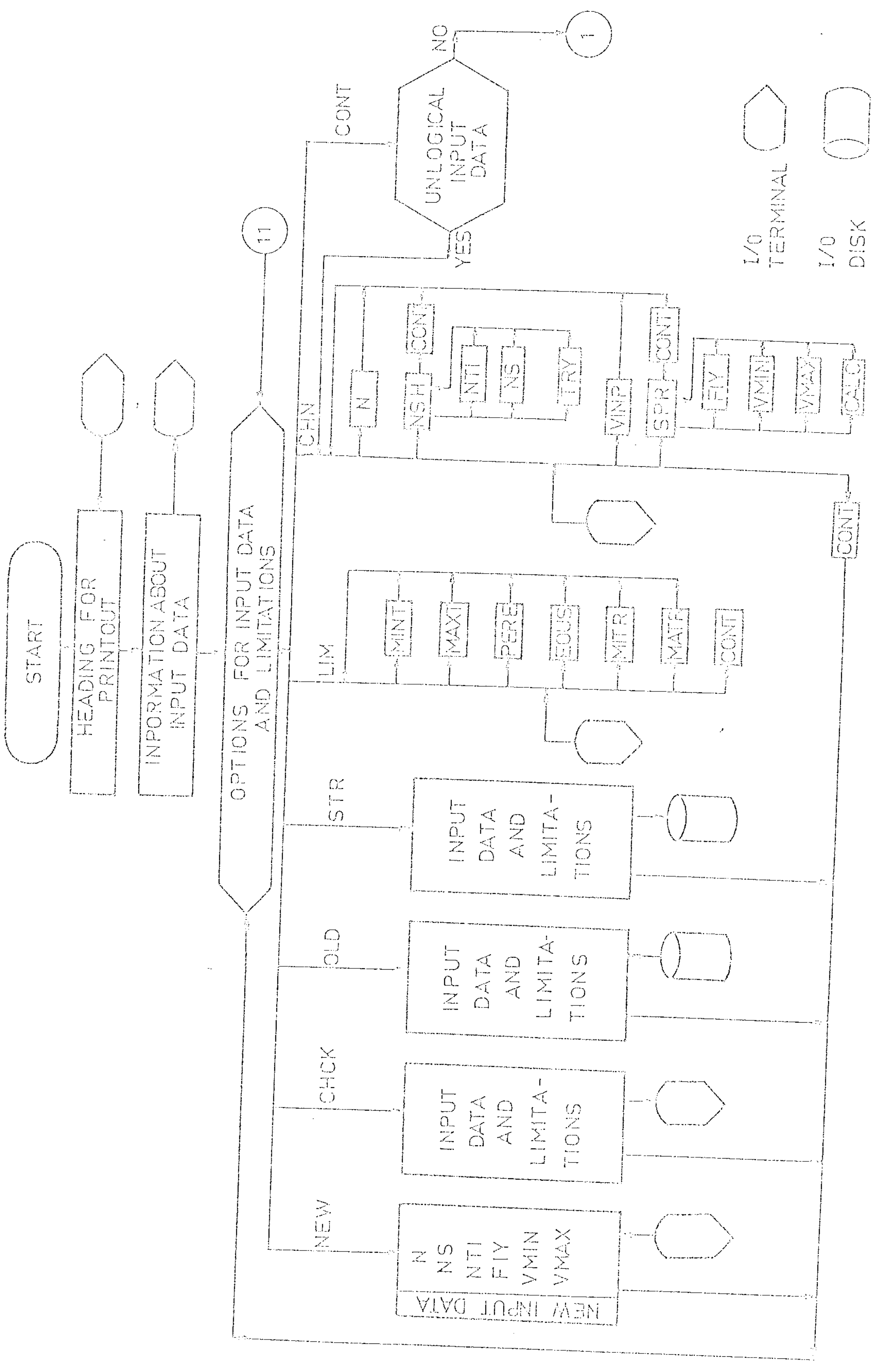
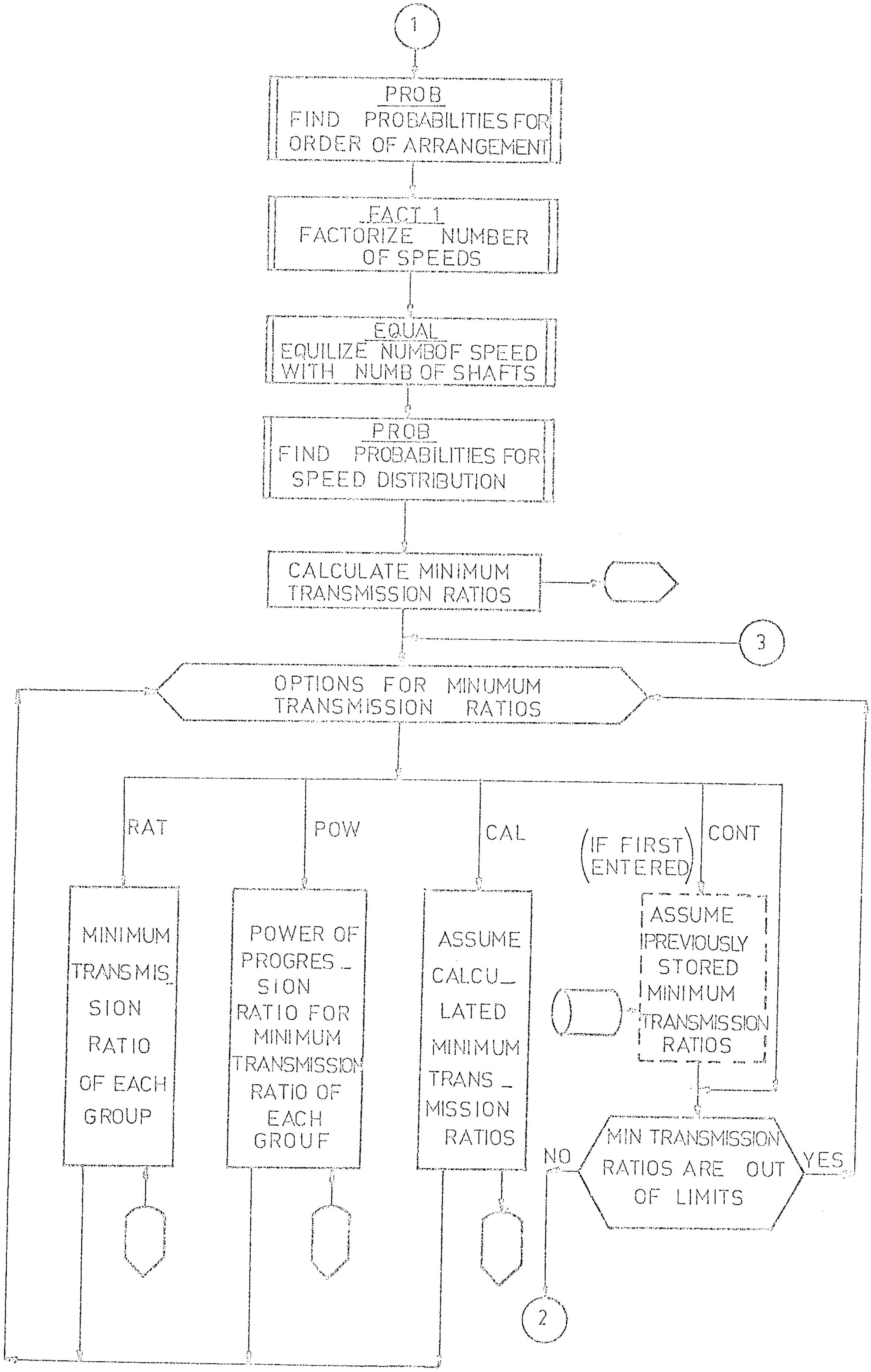
31- Tos FA4A, U, V Freze Katalogü

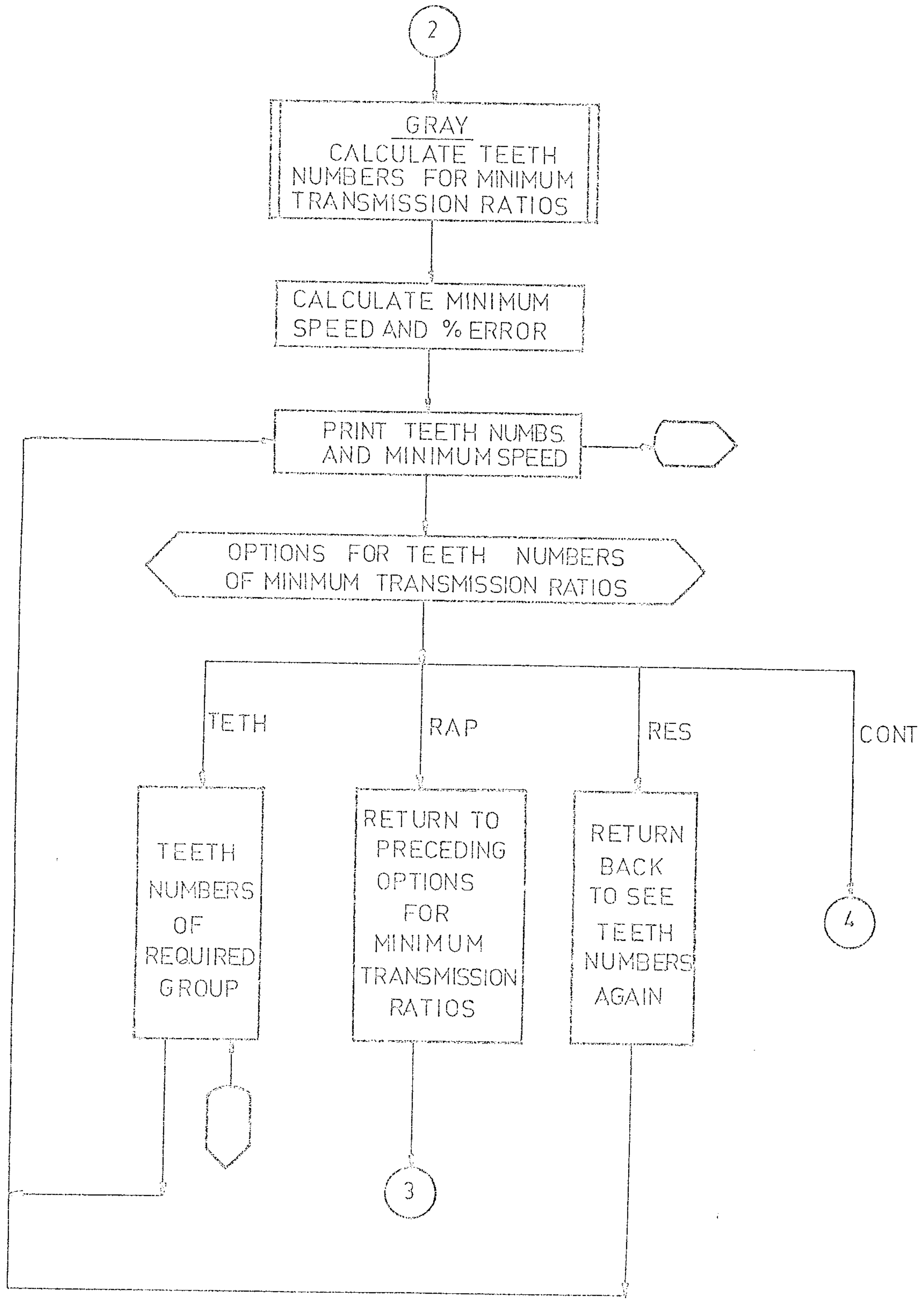
32- Ajax Torna Katalogü

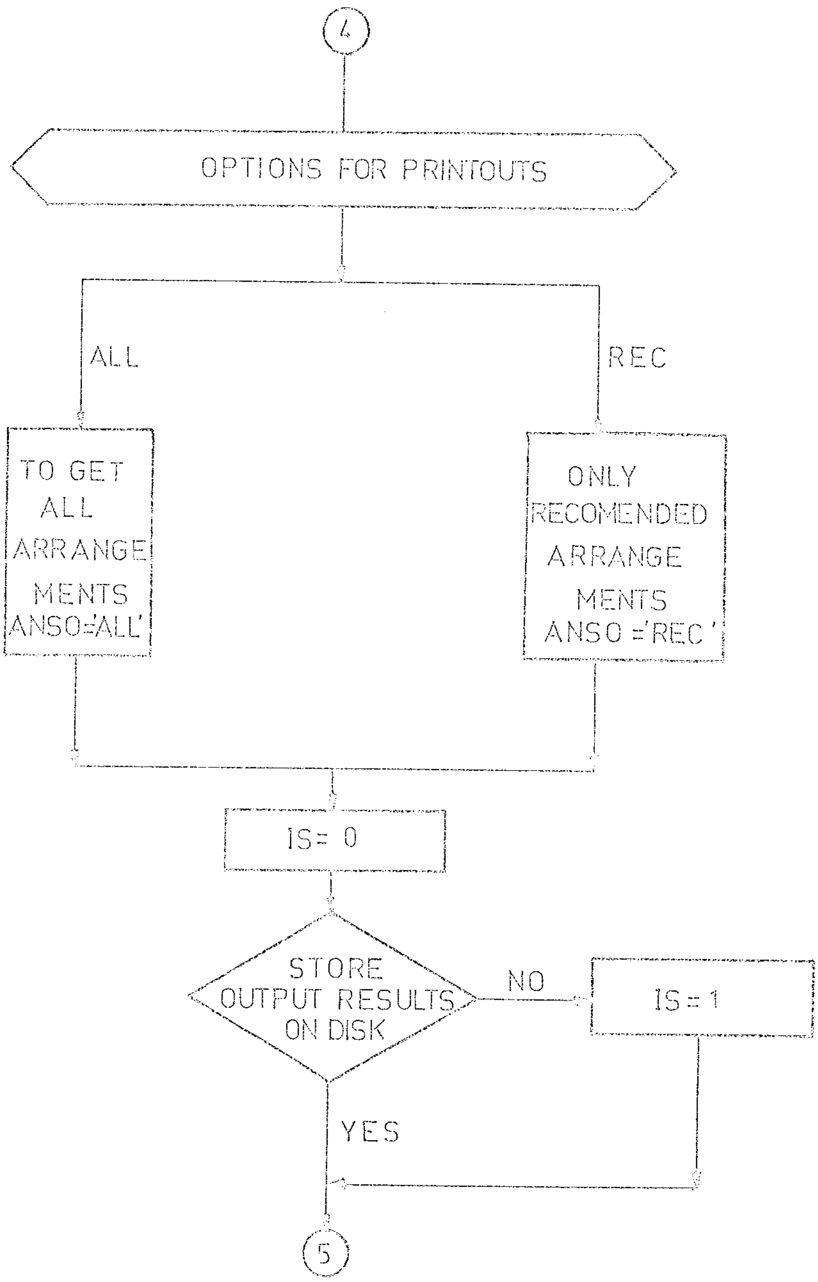
33- Enshu Milling Machines Katalog

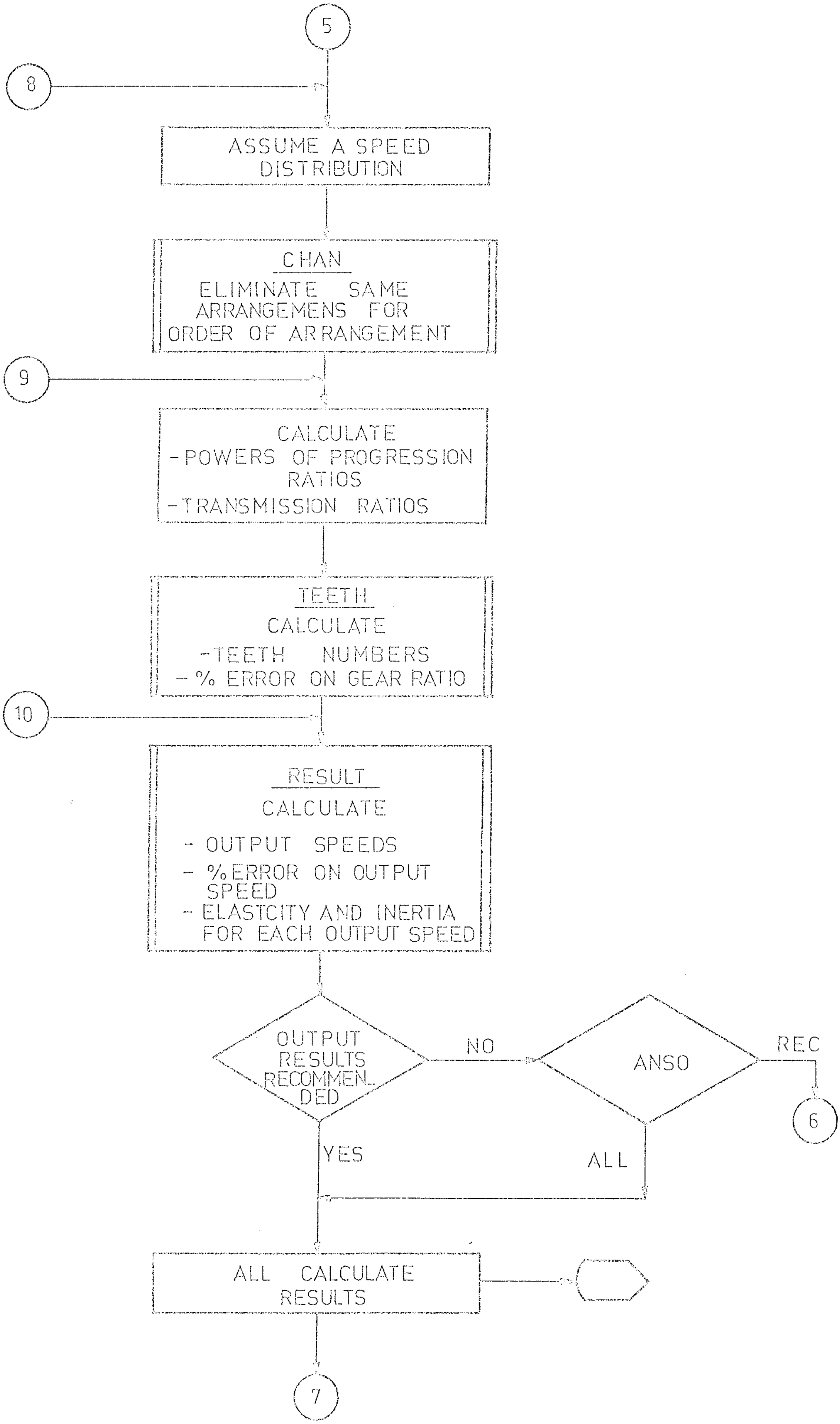
Appendix 5.1.a

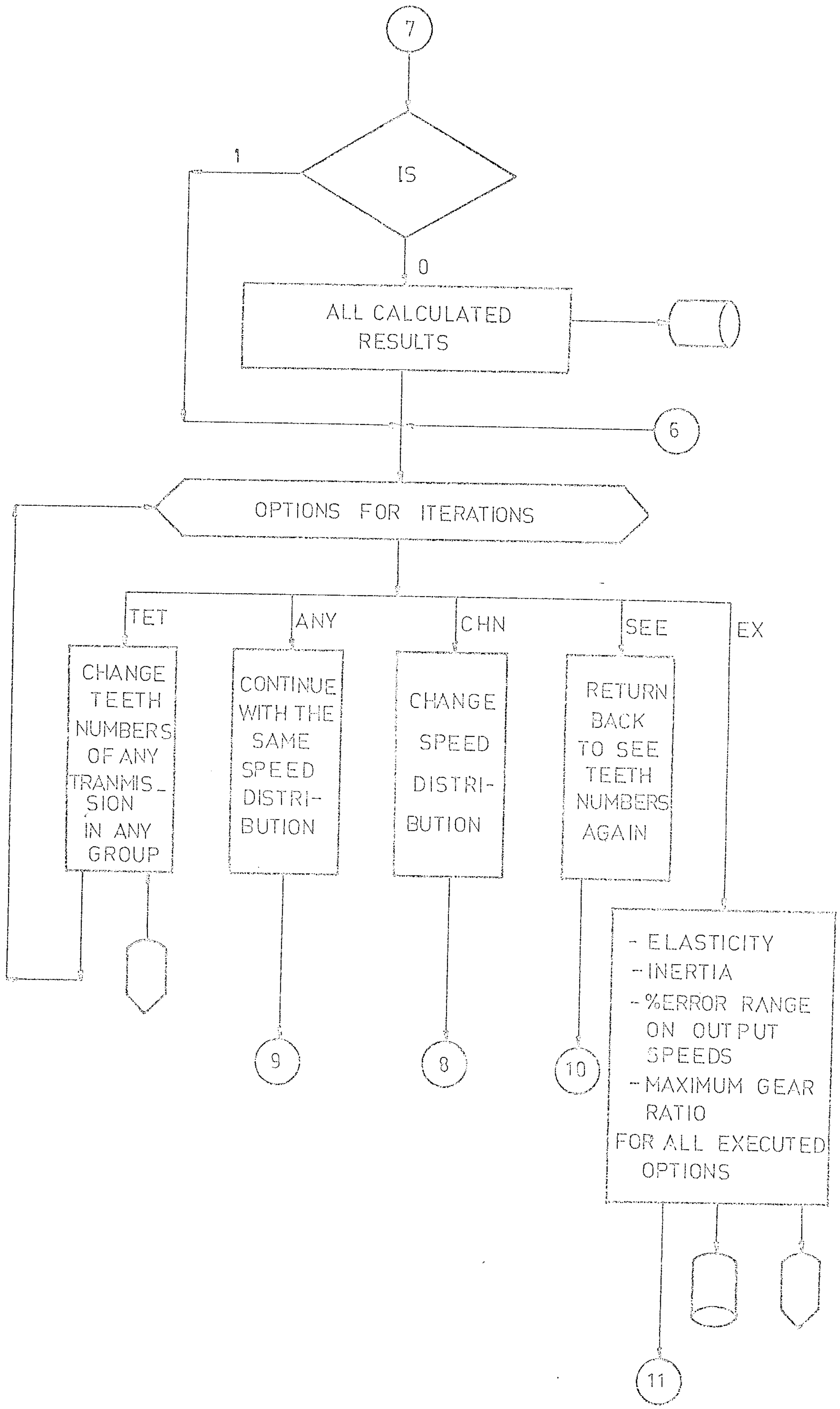


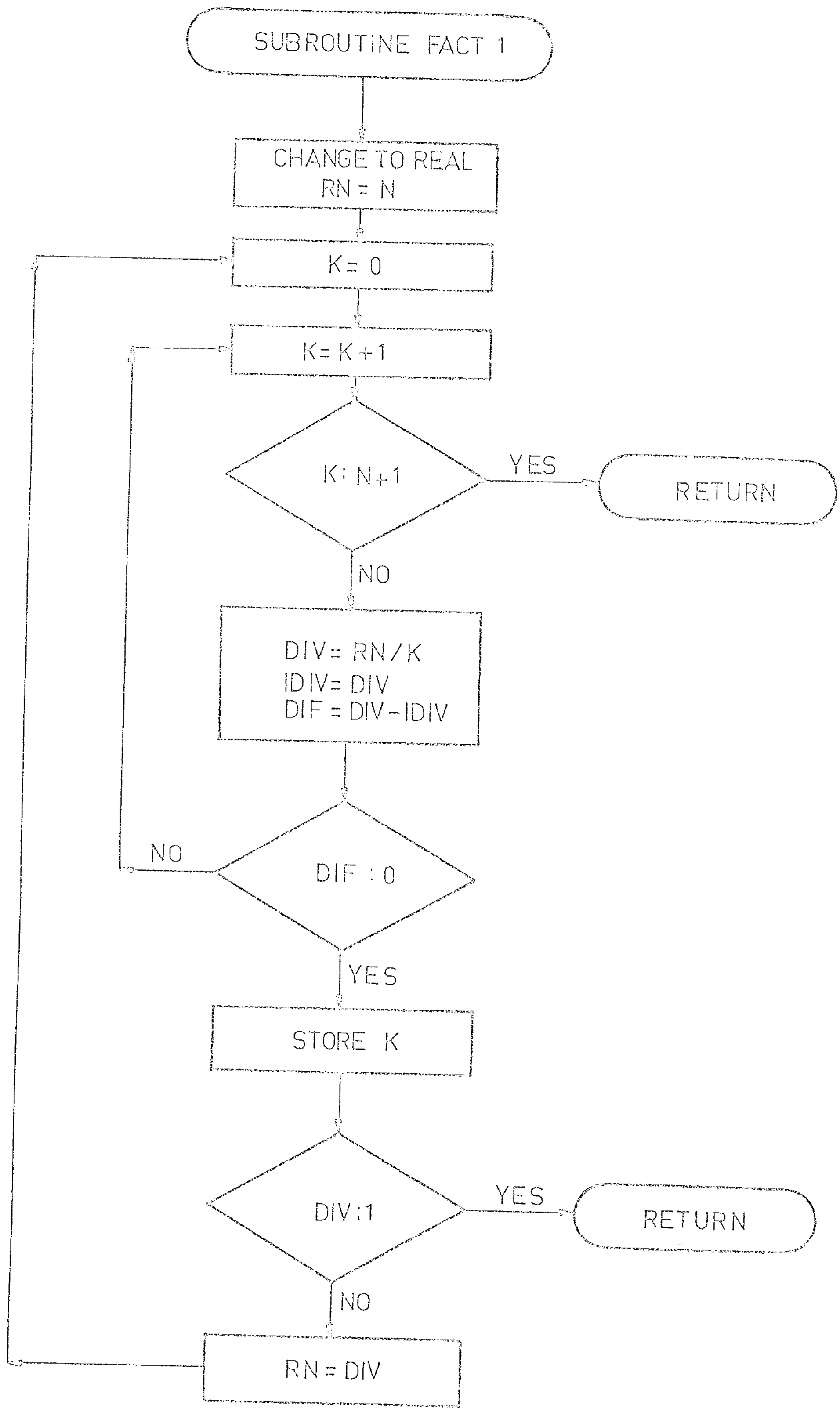

1
PROB
FIND PROBABILITIES FOR ORDER OF ARRANGEMENT
FACT 1
FACTORIZE NUMBER OF SPEEDS
EQUAL
EQUILIZE NUMBOF SPEED WITH NUMB OF SHAFTS
PROB
FIND PROBABILITIES FOR SPEED DISTRIBUTION
CALCULATE MINIMUM TRANSMISSION RATIOS
3
OPTIONS FOR MINUMUM TRANSMISSION RATIOS
RAT
POW
CAL
CONT
(IF FIRST ENTERED)
MINIMUM TRANSMIS_SION RATIO OF EACH GROUP
POWER OF PROGRES_SION RATIO FOR MINIMUM TRANSMISSION RATIO OF EACH GROUF
ASSUME CALCU_LATED MINIMUM TRANS_MISSION RATIOS
ASSUME PREVIOUSLY STORED MINIMUM TRANSMISSION RATIOS
MIN TRANSMISSION RATIOS ARE OUT OF LIMITS
NO
YES
2

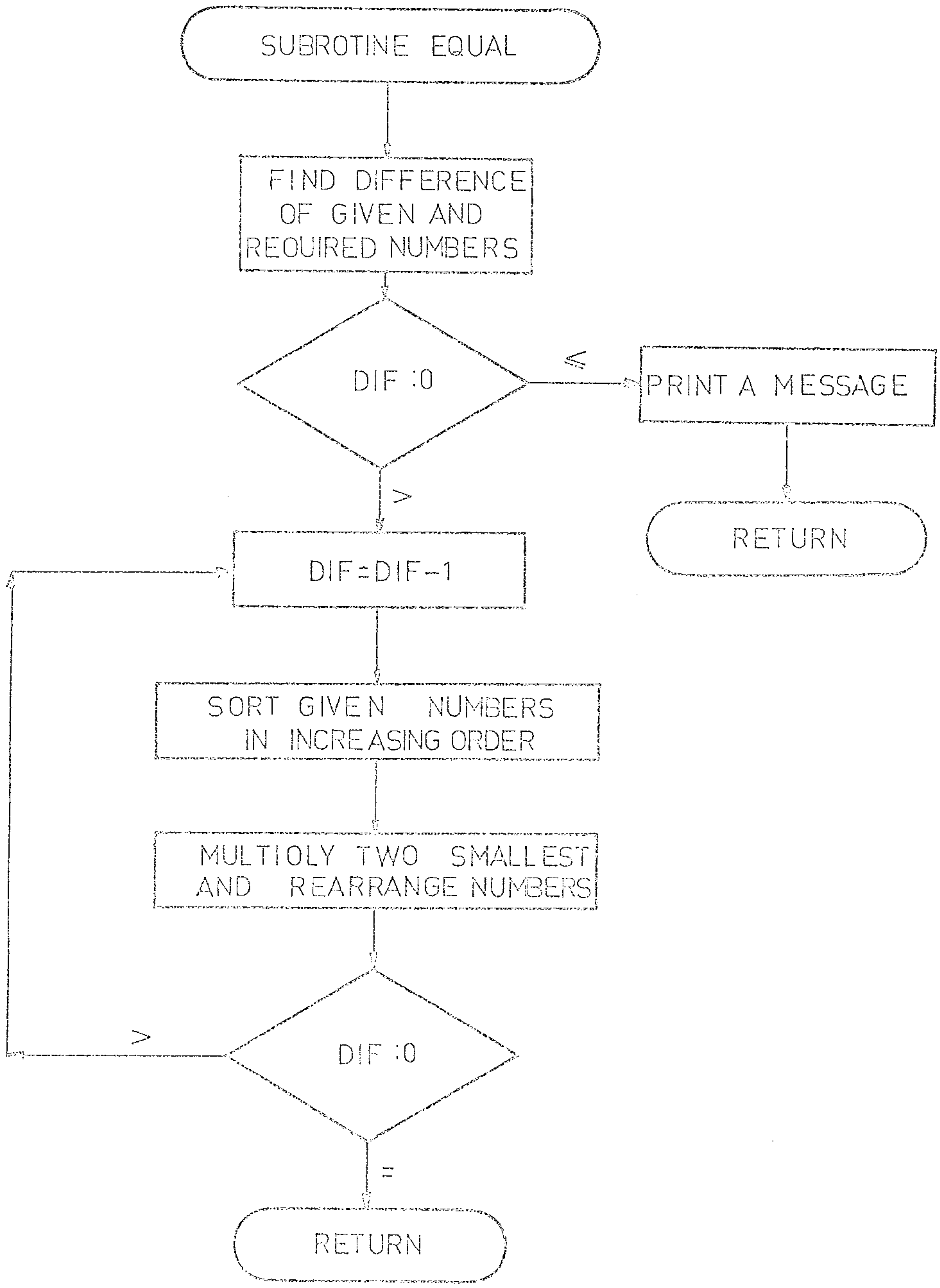

2
GRAY
CALCULATE TEETH NUMBERS FOR MINIMUM TRANSMISSION RATIOS
CALCULATE MINIMUM SPEED AND % ERROR
PRINT TEETH NUMBS. AND MINIMUM SPEED
OPTIONS FOR TEETH NUMBERS OF MINIMUM TRANSMISSION RATIOS
TETH
RAP
RES
CONT
TEETH NUMBERS OF REQUIRED GROUP
RETURN TO PRECEDING OPTIONS FOR MINIMUM TRANSMISSION RATIOS
RETURN BACK TO SEE TEETH NUMBERS AGAIN
4
3

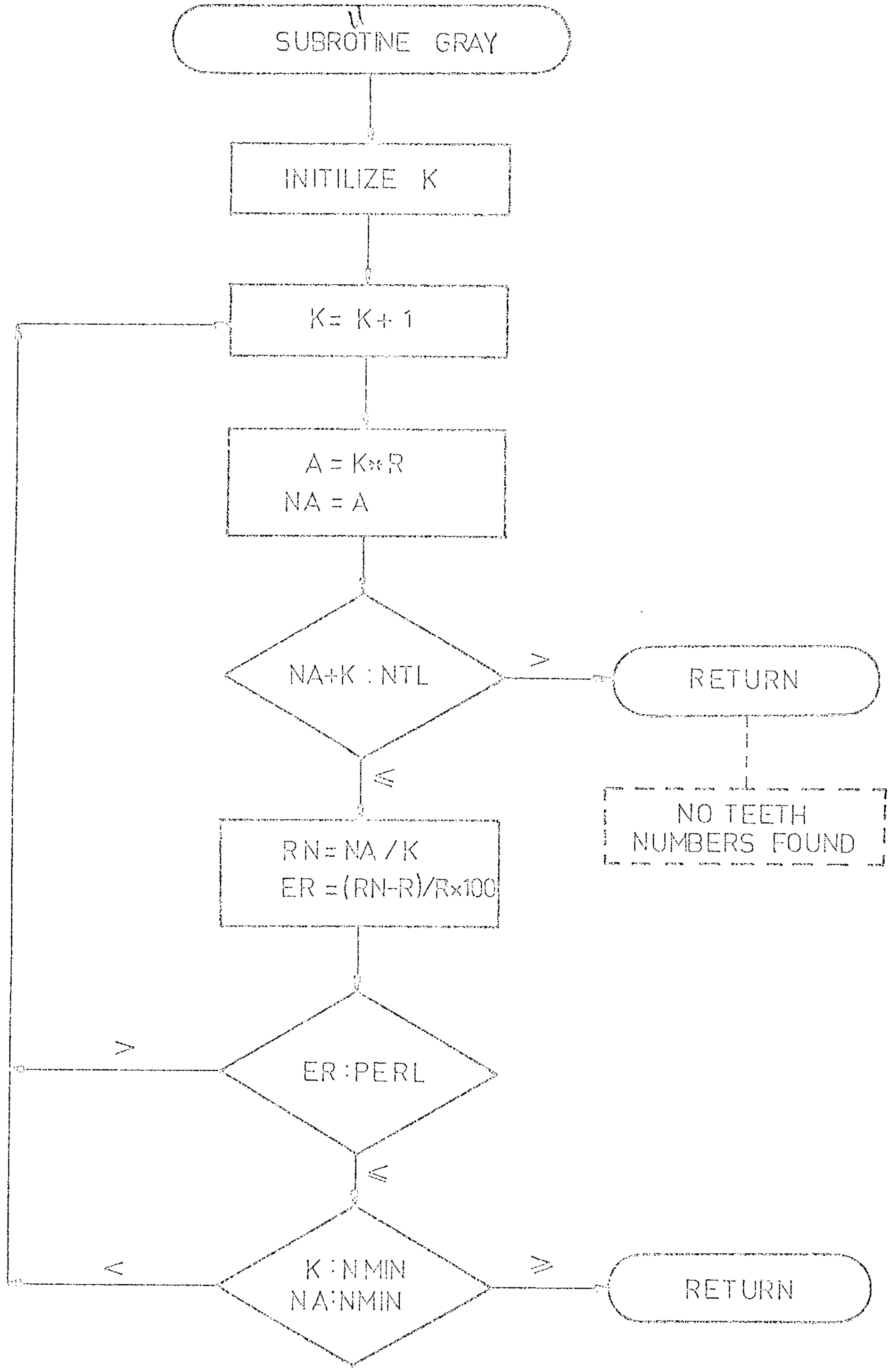

4
OPTIONS FOR PRINTOUTS
ALL
REC
TO GET ALL ARRANGE MENTS ANSO='ALL'
ONLY RECOMENDED ARRANGE MENTS ANSO ='REC'
IS= 0
STORE OUTPUT RESULTS ON DISK
NO
IS= 1
YES
5

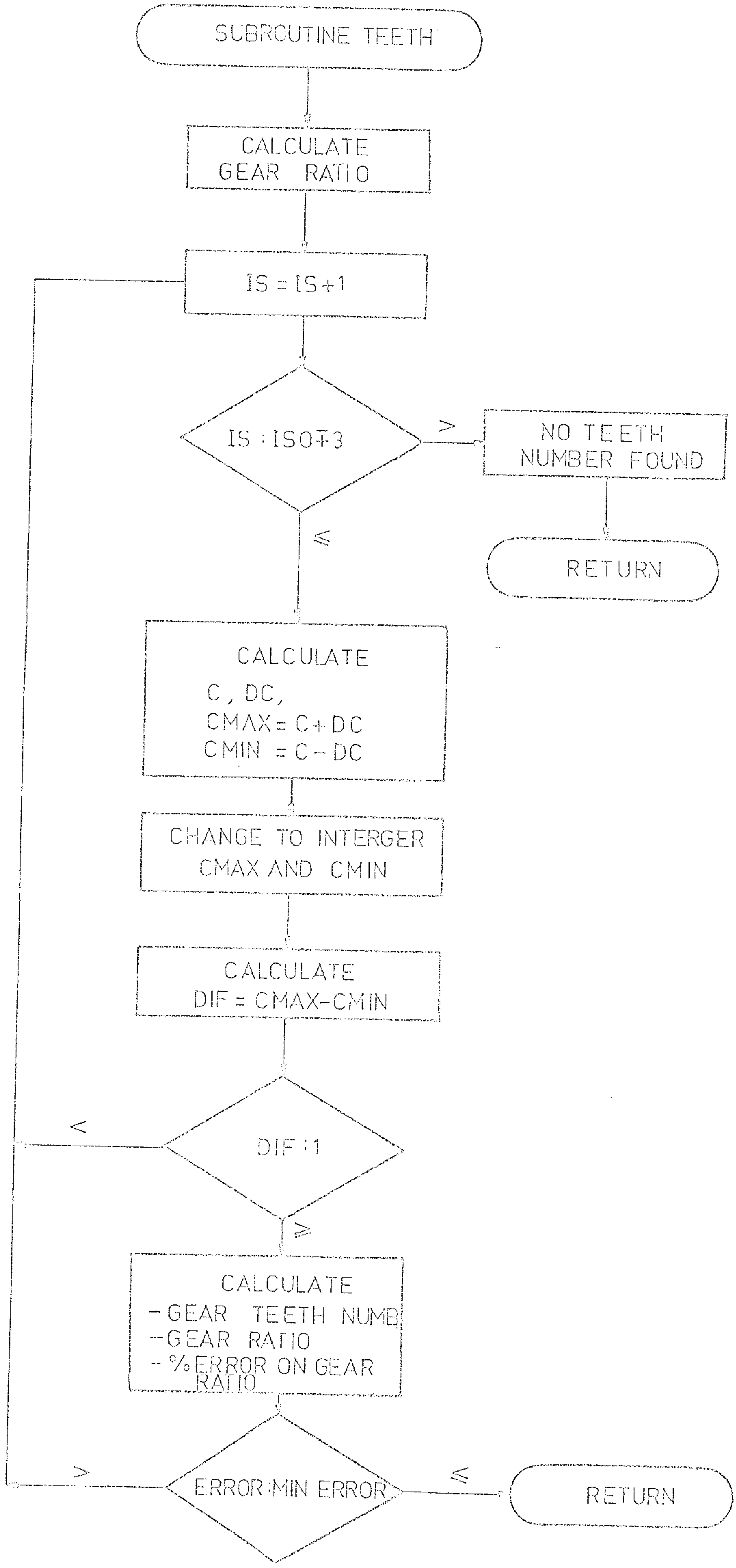

5
8
ASSUME A SPEED DISTRIBUTION
CHAN
ELIMINATE SAME ARRANGEMENS FOR ORDER OF ARRANGEMENT
9
CALCULATE
-POWERS OF PROGRESSION RATIOS
-TRANSMISSION RATIOS
TEETH
CALCULATE
-TEETH NUMBERS
-% ERROR ON GEAR RATIO
10
RESULT
CALCULATE
- OUTPUT SPEEDS
- % ERROR ON OUTPUT SPEED
- ELASTCITY AND INERTIA FOR EACH OUTPUT SPEED
OUTPUT RESULTS RECOMMEN_ DED
NO
ANSO
REC
6
YES
ALL
ALL CALCULATE RESULTS
7

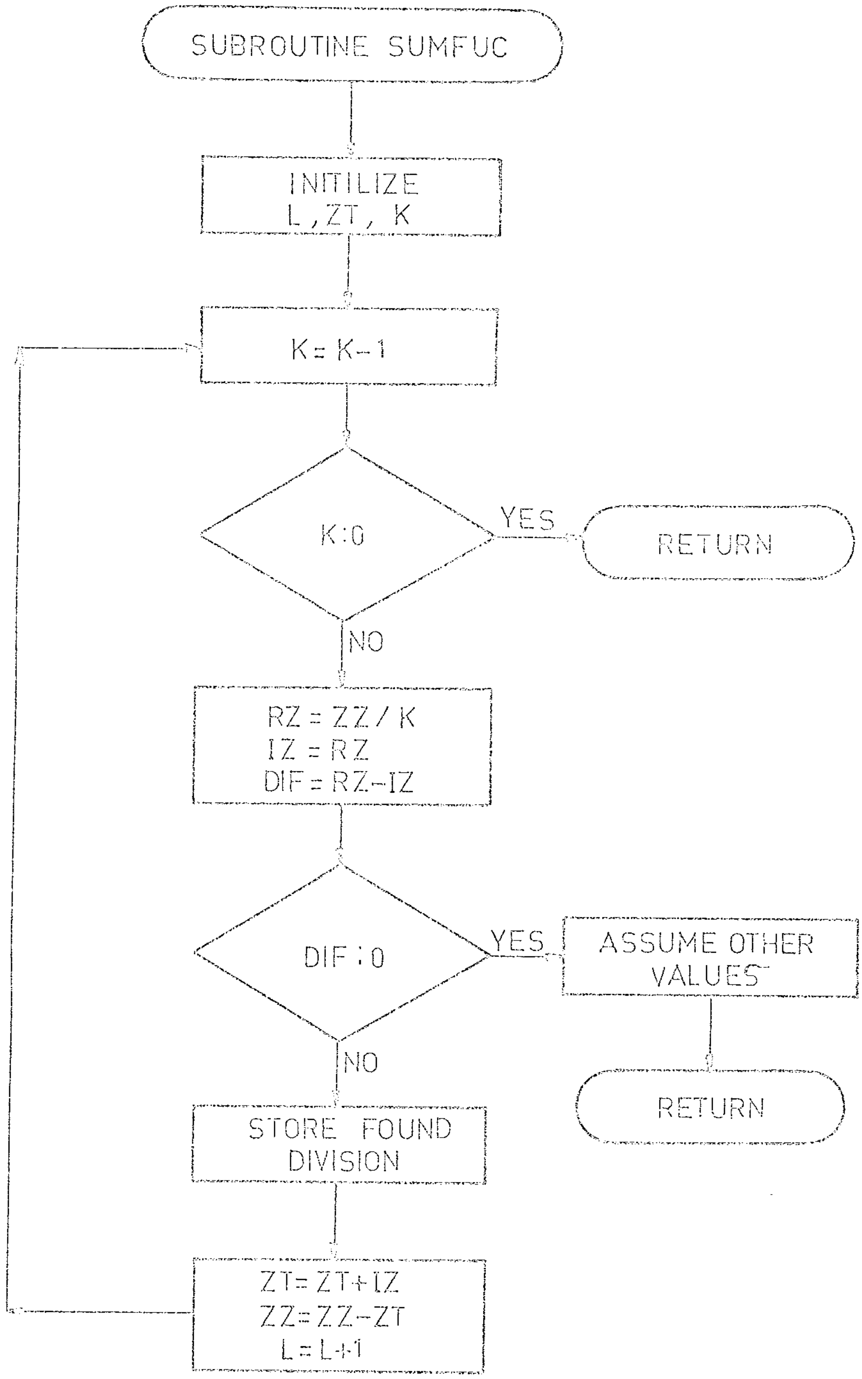

7
IS
1
0
ALL CALCULATED RESULTS
6
OPTIONS FOR ITERATIONS
TET
ANY
CHN
SEE
EX
CHANGE TEETH NUMBERS OF ANY TRANMIS_SION IN ANY GROUP
CONTINUE WITH THE SAME SPEED DISTRI-BUTION
CHANGE SPEED DISTRI-BUTION
RETURN BACK TO SEE TEETH NUMBERS AGAIN
- ELASTICITY
- INERTIA
- %ERROR RANGE ON OUTPUT SPEEDS
- MAXIMUM GEAR RATIO
FOR ALL EXECUTED OPTIONS
9
8
10
11

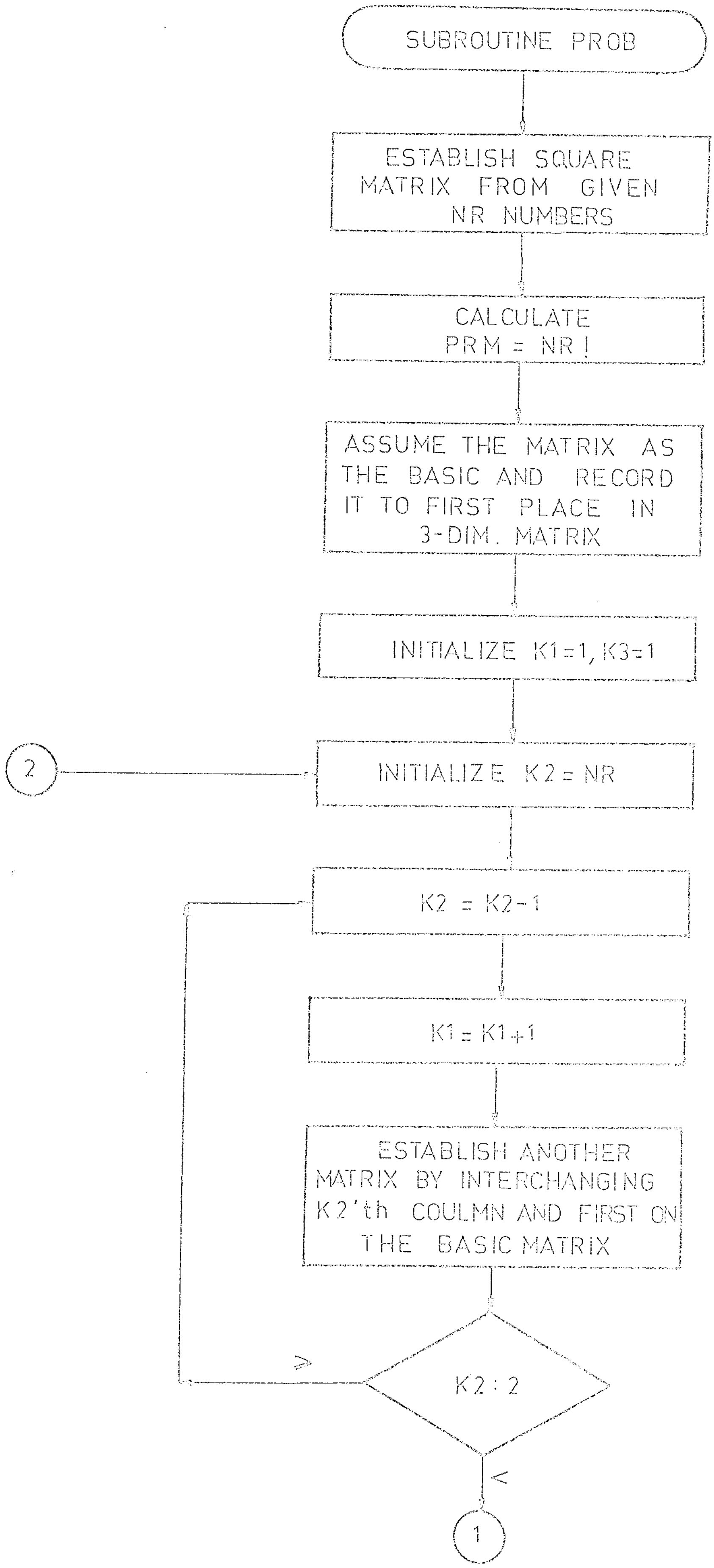

SUBROUTINE FACT 1
CHANGE TO REAL
RN = N
K= 0
K= K+1
K: N+1
YES
RETURN
NO
DIV= RN/K
IDIV= DIV
DIF = DIV-IDIV
NO
DIF : 0
YES
STORE K
DIV:1
YES
RETURN
NO
RN = DIV

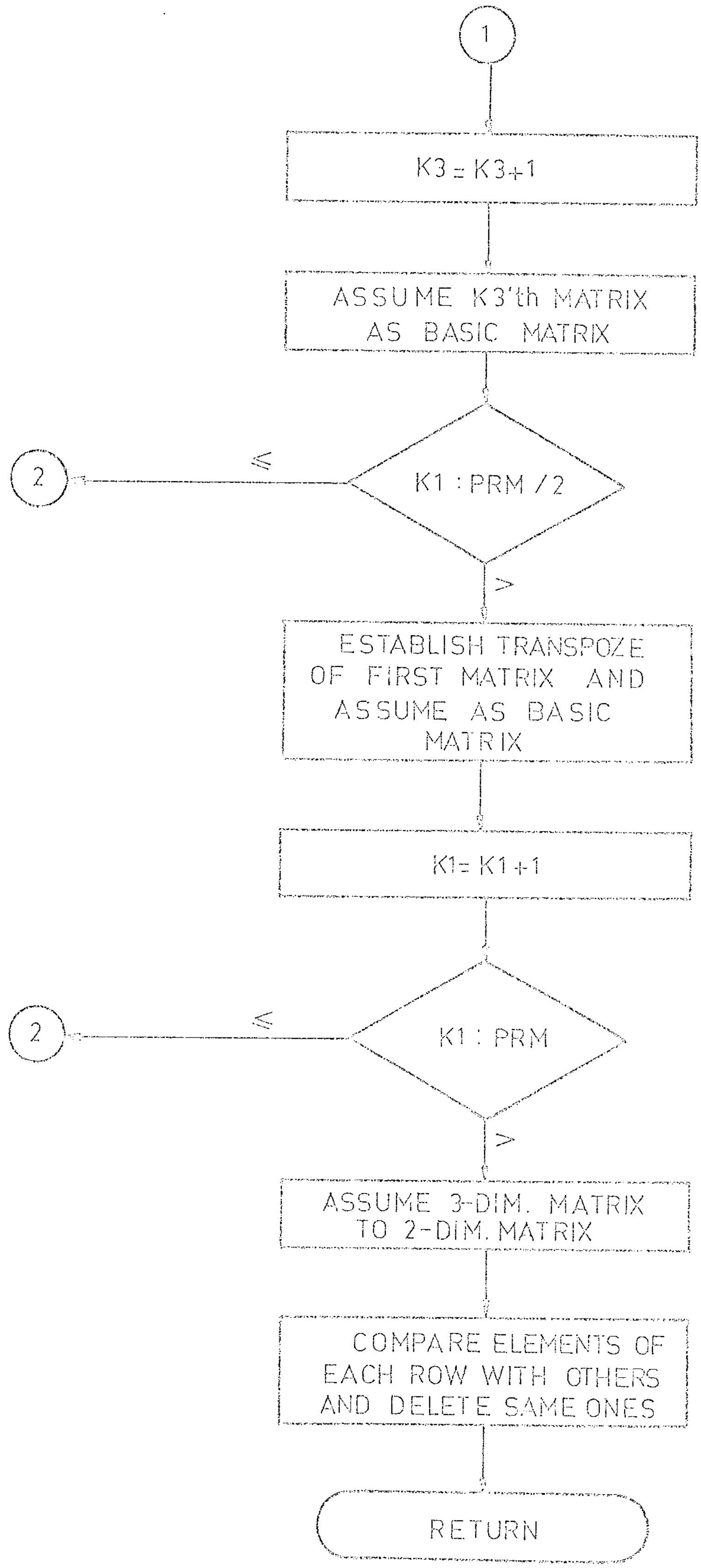

SUBROTINE EQUAL
FIND DIFFERENCE OF GIVEN AND REQUIRED NUMBERS
DIF :0
≤
PRINT A MESSAGE
RETURN
>
DIF=DIF-1
SORT GIVEN NUMBERS IN INCREASING ORDER
MULTIOLY TWO SMALLEST AND REARRANGE NUMBERS
>
DIF :0
=
RETURN

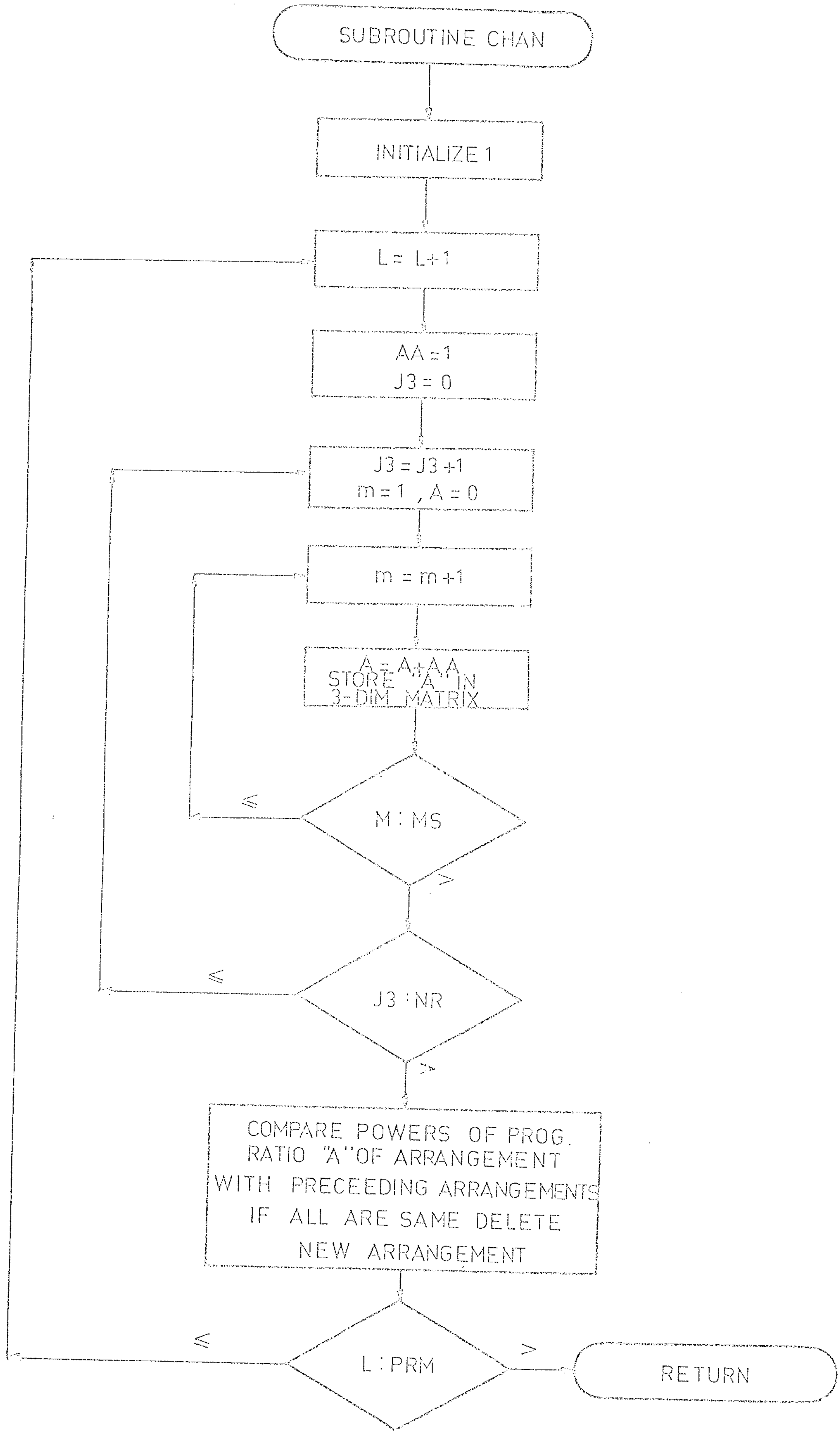

SUBROTINE GRAY
INITILIZE K
K= K+ 1
A = K*R
NA = A
NA+K : NTL
>
RETURN
NO TEETH
NUMBERS FOUND
≤
RN= NA / K
ER = (RN-R)/R×100
>
ER : PERL
≤
<
K : NMIN
NA : NMIN
≥
RETURN

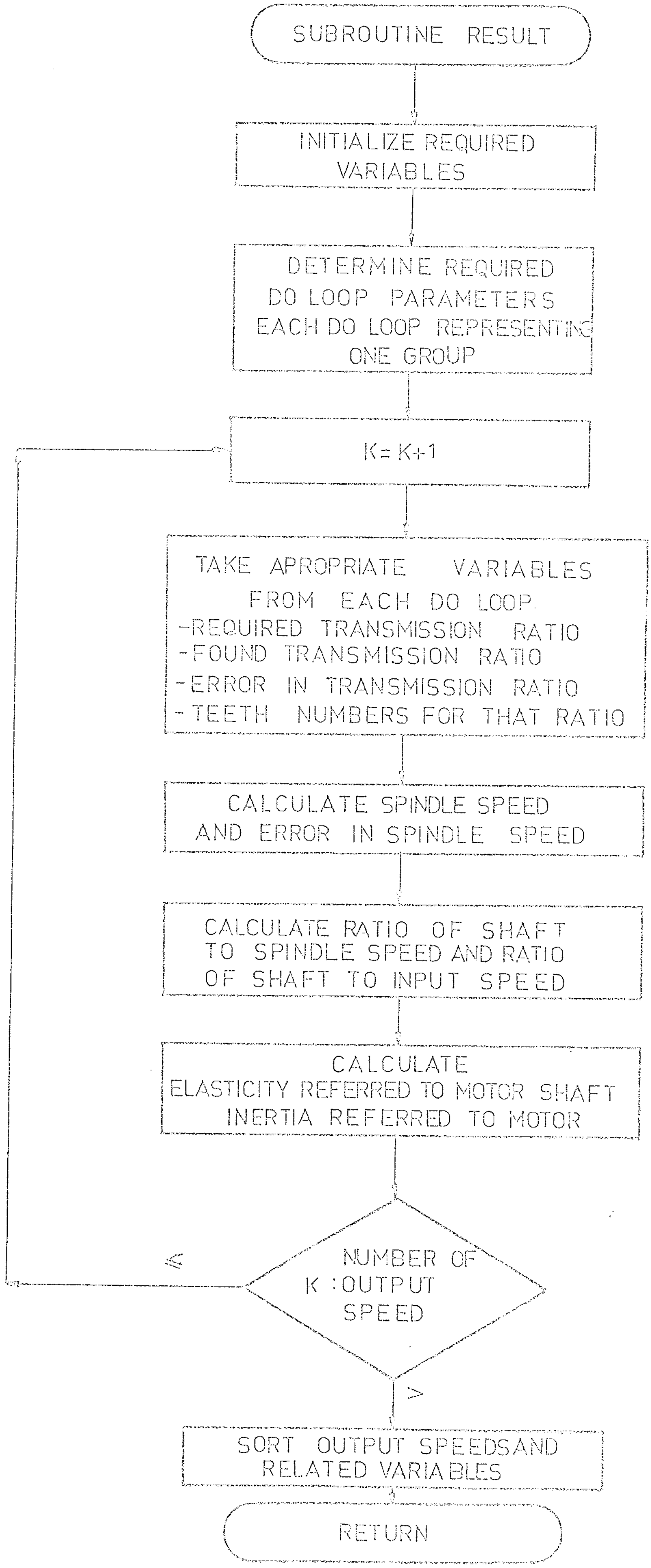

SUBROUTINE TEETH
CALCULATE GEAR RATIO
IS = IS+1
IS : ISO∓3
>
NO TEETH NUMBER FOUND
RETURN
≤
CALCULATE
C , DC,
CMAX = C+DC
CMIN = C−DC
CHANGE TO INTERGER CMAX AND CMIN
CALCULATE DIF = CMAX−CMIN
<
DIF : 1
≥
CALCULATE
−GEAR TEETH NUMB
−GEAR RATIO
− % ERROR ON GEAR RATIO
>
ERROR:MIN ERROR
≤
RETURN


SUBROUTINE SUMFUC
INITILIZE
L ,ZT , K
K = K-1
K:0
YES
RETURN
NO
RZ = ZZ / K
IZ = RZ
DIF = RZ-IZ
DIF : 0
YES
ASSUME OTHER
VALUES
RETURN
NO
STORE FOUND
DIVISION
ZT= ZT+IZ
ZZ= ZZ-ZT
L = L+1


SUBROUTINE PROB
ESTABLISH SQUARE MATRIX FROM GIVEN NR NUMBERS
CALCULATE PRM = NR !
ASSUME THE MATRIX AS THE BASIC AND RECORD IT TO FIRST PLACE IN 3-DIM. MATRIX
INITIALIZE K1=1, K3=1
2
INITIALIZE K2 = NR
K2 = K2-1
K1 = K1+1
ESTABLISH ANOTHER MATRIX BY INTERCHANGING K2'th COULMN AND FIRST ON THE BASIC MATRIX
K2 : 2
>
<
1


1
K3 = K3+1
ASSUME K3'th MATRIX AS BASIC MATRIX
K1 : PRM /2
≤
2
>
ESTABLISH TRANSPOZE OF FIRST MATRIX AND ASSUME AS BASIC MATRIX
K1= K1+1
K1 : PRM
≤
2
>
ASSUME 3-DIM. MATRIX TO 2-DIM. MATRIX
COMPARE ELEMENTS OF EACH ROW WITH OTHERS AND DELETE SAME ONES
RETURN


SUBROUTINE CHAN
INITIALIZE 1
L= L+1
AA = 1
J3 = 0
J3 = J3 +1
m = 1 , A = 0
m = m +1
A = A+AA
STORE "A" IN
3-DIM MATRIX
M : MS
≤
>
J3 : NR
≤
>
COMPARE POWERS OF PROG.
RATIO "A" OF ARRANGEMENT
WITH PRECEEDING ARRANGEMENTS
IF ALL ARE SAME DELETE
NEW ARRANGEMENT
L : PRM
≤
>
RETURN


SUBROUTINE RESULT
INITIALIZE REQUIRED VARIABLES
DETERMINE REQUIRED DO LOOP PARAMETERS EACH DO LOOP REPRESENTING ONE GROUP
K= K+1
TAKE APROPRIATE VARIABLES FROM EACH DO LOOP.
-REQUIRED TRANSMISSION RATIO
-FOUND TRANSMISSION RATIO
-ERROR IN TRANSMISSION RATIO
-TEETH NUMBERS FOR THAT RATIO
CALCULATE SPINDLE SPEED AND ERROR IN SPINDLE SPEED
CALCULATE RATIO OF SHAFT TO SPINDLE SPEED AND RATIO OF SHAFT TO INPUT SPEED
CALCULATE
ELASTICITY REFERRED TO MOTOR SHAFT
INERTIA REFERRED TO MOTOR
≤
NUMBER OF
K : OUTPUT
SPEED
>
SORT OUTPUT SPEEDSAND RELATED VARIABLES
RETURN

Appendix 5.1.b

```
***************************************************************
*                                                             *
*   ALL POSSIBLE SOLUTIONS                                    *
*   FOR THE KINEMATIC ARRANGEMENT DIAGRAM                     *
*   OF GEARBOXES FOR MACHINE TOOLS                            *
*                                                             *
*                                                             *
*                                                             *
*                                                             *
*                               BY  SEDAT KARSLI              *
*                                                             *
*                                   SEPTEMBER 1985            *
*                                                             *
***************************************************************
```

```
                E X A M P L E

          FOR THE ILLUSTRATION OF

        THE FLOW OF COMPUTER PROGRAM




REQUIRED MINIMUM INPUT DATA FOR EXECUTION

     *   NUMBER OF SPEEDS

TWO OF THE FOLLOWING THREE

     *   MINIMUM SPEED
     *   MAXIMUM SPEED
     *   PROGRESSION RATIO (STANDARD VALUES GIVEN IN PROGRAM)

THE FOLLOWING INPUT DATA CAN BE DECIDED DURING EXECUTION

     *   NUMBER OF TOTAL SHAFTS
     *   NUMBER OF SHAFTS WITH ONE TRANSMISSION
     *   MOTOR SPEED (STANDARD VALUES GIVEN IN PROGRAM)
     *   MINIMUM TRANSMISSION RATIOS


USE  ?  TO SEE THE DEAILS OF OPTIONS (IF THERE IS)

CLEAR TERMINAL BEFORE ENTERING ANY OPTION (FOR BETTER VISUAL)

TO SEE THE OPTIONS AGAIN PRES ANY KEY
```

```
OPTIONS AVAILABLE

 NEW  : FOR NEW INPUT DATA
 CHN  : TO CHANGE DATA
 OLD  : FOR OLD INPUT DATA
 STR  : TO STORE PRESENT INPUT DATA
 LIM  : FOR LIMITATIONS
 CHCK : TO CHECK INPUT DATA
 CONT : TO CONTIUE
 EX   : TO EXIT

OPTION :NEW

   NUMBER OF SPEEDS  :  8


OPTIONS FOR NUMBER OF SHAFTS

 ?    : FOR HELP
 TNL  : FOR NUMBER OF SHAFTS WITH 1 TRANSMISSION
 TNS  : FOR TOTAL NUMBER OF SHAFTS
 TRY  : TO TRY WITH THE NUMBERS OF SHAFTS
 CONT : TO CONTINUE

OPTION FOR NUMBER OF SHAFTS  : ?

   ENTER * TNS * IF YOU KNOW THE TOTAL NUMBER OF
                 SHAFTS TO BE USED IN THE GEARBOX

   ENTER * TNL * TO PRINT NUMBER OF SHAFTS
                 WHICH WILL CARRY ONLY ONE TRANSMISSION

   ENTER * TRY * IF YOU WANT TO SEE MAXIMUM NUMBER OF GEARS
                 TO BE USED STARTING WITH 2 SHAFTS AND UP TO 10 SHAFTS
   YOU HAVE TO INPUT AT LEAST TOTAL NUMBER OF SHAFTS
   WHICH IS EQUAL OR GRATER THAN 2

OPTION FOR NUMBER OF SHAFTS  : TNL
   NUMBER OF SHAFTS WITH 1 TRANSMISSION :  0

OPTION FOR NUMBER OF SHAFTS  : TNS
   NUMBER OF SHAFTS  :  3

OPTION FOR NUMBER OF SHAFTS  : CONT
   DO YOU KNOW INPUT SPEED ?  : NO
   INPUT MOTOR SPEEDS ARE STANDARDIZED AS

       3000_2000_1500_1200_1000_750_600_500_375_300  REV/MIN

BUT TESE MOTOR SPEEDS ARE  REDUCED UNDER THE LOAD APROXIMATELY
BETWEEN 6.5% AND 13.5% .DURING CALCULATION FOR MACHINE TOOL
GEAR_BOX ON LOAD SPEEDS ARE USED. THESE ON LOAD SPEEDS ARE;

       2800_1800_1400_1120_900_710_560_450_355_280  REV/MIN

  GENERALLY FOR MACHIN TOOL DRIVES A.C. ELECTRICAL MOTORS WITH
SPEEDS OF ** 1400 ** 900 ** REV/MIN ARE USED.

PRINT THE VALUE OF INPUT SPEED AS REQUIRED
   INPUT SPEED    :    900.00
```

OPTIONS FOR SPEED RANGE

? : FOR HELP
FIY : FOR PROGRESSION RATIO
VMIN : FOR MINIMUM SPEED
VMAX : FOR MAXIMUM SPEED
CALC : FOR CALCULATIONS OF SPEED RANGE (OPTIONAL)
CONT : TO CONTINUE

OPTION FOR SPEED RANGE : ?

FOR SPEED RANGE YOU MUST ENTER AT LEAST 2 OF THE OPTIONS
FIY
VMIN
VMAX
USE CALC TO SEE ALL OF THE OUTPUT SPEEDS (IF NEEDED)
MINIMUM OR MAXIMUM SPEED IS DETERMINED
ACCORDING TO THE CUTTING SPEED REQUIREMENTS
YOU ARE EXPECTED TO KNOW AT LEAST ONE OF THEM

OPTION FOR SPEED RANGE : FIY
DO YOU KNOW STANDARD PROGRESSION RATIO ? : NO

****

RANGE RATIOS

****

NUMBER OF SPEEDS	PROGRESSION RATIOS						
	1.06	1.12	1.26	1.41	1.58	1.78	2.00
4	1.19	1.40	2.00	2.80	3.94	5.64	8
8	1.50	2.21	5.04	11.03	24.58	56.62	128
12	1.90	3.48	12.71	43.79	153.17	568.36	2048
16	2.40	5.47	32.03	173.09	954.68	5705.6	
20	3.03	8.61	80.73	684.17	5949.6		
24	3.82	13.55	203.48	2704.2			
28	4.82	21.32	512.87	10688			
32	6.09	33.56	1292.7				
36	7.67	52.80	3258.1				
40	9.70	83.08	8212.0				
44	12.25	130.73					
48	15.47	205.71					
52	19.53	323.68					
56	24.65	509.32					

*** RECOMMENDED VALUES ***

USE FIY=1.26 OR 1.41 FOR GENERAL PURPOSE MACHINE TOOLS

USE FIY=1.12 OR 1.26 FOR AUTOMATIC OR SEMIAUTOMATIC
MACHINE TOOLS INTENDED FOR MASS PRODUCTION

USE FIY=1.58 OR 1.78 FOR SMALL MACHINE TOOLS
TO ACCOMMODATE SMALL WORK DIAMETERS

USE FIY=1.26 , 1.12 OR 1.06 FOR HEAVY MACHINE TOOLS

****

```
   PROGRESSION RATIO :        1.41

OPTION FOR SPEED RANGE  : VMIN
   MINIMUM SPEED     :     112.50

OPTION FOR SPEED RANGE  : CALC

   THEORETICAL OUTPUT SPEEDS WILL BE;
112.50 _    159.62 _    223.65 _    315.36 _    444.66 _

626.97 _    884.03 _  1246.48 _

   MINIMUM SPEED WILL BE ASSUMED TO BE  114.53
   IF YOU STRICTLY REQUIRE MINIMUM SPEED TO BE     112.50
   YOU HAVE TO INPUT MINIMUM TRANSMISSION RATIOS ACCORDINGLY
   AT A LATER STAGE

OPTION FOR SPEED RANGE  : CONT

OPTION :CHCK


   INPUT VALUES

     NUMBER OF SPEEDS          =     8
     NUMBER OF SHAFTS          =     3
     SHAFTS WITH 1 TRANS.      =     0
     INPUT SPEED               =   900.00
     MINIMUM SPEED             =   112.50
     MAXIMUM SPEED             =  1246.48
     PROGRESSION RATIO         =      1.4100

   LIMITATIONS

     MINIMUM TEETH NUMBER     =    18
     TOTAL TEETH NUMBER       =   120
     % ERROR ON GEAR RATIO    = \+ 1.00 %
     % ERROR ON OUTPUT SPEED= \+ 2.00 %
     MINIMUM TRANS. RATIO     =     0.25
     MAXIMUM TRANS. RATIO     =     2.00


   IF INPUT VALUES AND LIMITATIONS ARE CORRECT, CONTINUE
   OTHERWISE CORRECT INPUT DATA OR LIMITATIONS

OPTION :CIN


OPTIONS FOR INPUT VALUES

 NSP  : FOR NUMBER OF SPEEDS
 NSH  : FOR NUMBER OF SHAFTS
 SPI  : FOR INPUT SPEED
 SPR  : FOR SPEED RANGE
 CONT : TO CONTINUE


OPTION FOR INPUT VALUES : NSP

   NUMBER OF SPEEDS  :  6

OPTION FOR INPUT VALUES : SPR
```

```
OPTION FOR SPEED RANGE  : CALC

   THEORETICAL OUTPUT SPEEDS WILL BE;
112.50 _    158.62 _    223.66 _    315.36 _    444.66 _

626.97 _

   MINIMUM SPEED WILL BE ASSUMED TO BE  114.53
   IF YOU STRICTLY REQUIRE MINIMUM SPEED TO BE     112.50
   YOU HAVE TO INPUT MINIMUM TRANSMISSION RATIOS ACCORDINGLY
   AT A LATER STAGE

OPTION FOR SPEED RANGE  : CONT

OPTION FOR INPUT VALUES : CONT

OPTION :LIM


OPTIONS FOR LIMITATIONS

 ?    : FOR HELP
 MINT : FOR MINIMUM TEETH NUMBER
 MAXT : FOR MAXIMUM TOTAL TEETH NUMBER
 PERE : FOR PERCENTAGE ERROR ON GEAR RATIO
 ECUS : FOR ABS. PERCENTAGE ERROR ON OUTPUT SPEED
 MITR : FOR MINIMUM TRANSMISSION RATIO
 MATR : FOR MAXIMUM TRANSMISSION RATIO
 CONT : TO CONTINUE

OPTION FOR LIMITATIONS : MATR
   MAXIMUM TRANSMISSION RATIO   :    2.40

OPTION FOR LIMITATIONS : CONT

OPTION :CHCK


   INPUT VALUES

     NUMBER OF SPEEDS           =     6
     NUMBER OF SHAFTS           =     3
     SHAFTS WITH 1 TRANS.       =     0
     INPUT SPEED                =   900.00
     MINIMUM SPEED              =   112.50
     MAXIMUM SPEED              =   626.97
     PROGRESSION RATIO          =     1.4100

   LIMITATIONS

     MINIMUM TEETH NUMBER       =    18
     TOTAL TEETH NUMBER         =   120
     % ERROR ON GEAR RATIO      = \+ 1.00 %
     % ERROR ON OUTPUT SPEED    = \+ 2.00 %
     MINIMUM TRANS. RATIO       =     0.25
     MAXIMUM TRANS. RATIO       =     2.40


IF INPUT VALUES AND LIMITATIONS ARE CORRECT, CONTINUE
OTHERWISE CORRECT INPUT DATA OR LIMITATIONS
```

```
OPTION :STA

   STORED INPUT VALUES

     NUMBER OF SPEEDS          =     6
     NUMBER OF SHAFTS          =     3
     SHAFTS WITH 1 TRANS.      =     0
     INPUT SPEED               =   900.00
     MINIMUM SPEED             =   112.50
     MAXIMUM SPEED             =   626.97
     PROGRESSION RATIO         =     1.4100

   STORED LIMITATIONS

     MINIMUM TEETH NUMBER      =    18
     TOTAL TEETH NUMBER        =   120
     % ERROR ON GEAR RATIO     = \+ 1.00 %
     % ERROR ON OUTPUT SPEED= \+ 2.00 %
     MINIMUM TRANS. RATIO      =     0.25
     MAXIMUM TRANS. RATIO      =     2.40

   STORED MINIMUM TRANSMISSION RATIOS

     1.00000000 = 1.000000 * 1.000000 * 1.000000 *

OPTION :CONT

   CALCULATED MINIMUM TRANSMISSION RATIOS ARE

     0.12725818 = 0.502993 * 0.253002 *


OPTIONS FOR MINIMUM TRANSMISSION RATIOS

 ?    : FOR HELP
 RAT  : TO PRINT NEW TRANSMISSION RATIOS
 POW  : TO PRINT POWERS OF PROGRESSION RATIO
        TO FIND NEW TRANSMISSION RATIOS
 CAL  : TO USE CALCULATED MINIMUM TRANSMISSION RATIOS
 CONT : TO CONTINUE
        TO USE OLD TRANSMISSION RATIOS CONTINUE DIRECTLY


OPTION FOR MINIMUM TRANSMISSION RATIO  : TAR

ENTER CORRECT WORDS FOR THE OPTION


OPTION FOR MINIMUM TRANSMISSION RATIO  : RAT

PRINT VALUES OF MINIMUM TRANSMISSION RATIOS ONE BY ONE
   BY A DECREASING ORDER FROM FIRST TO LAST SHAFT
   YOU WILL PRINT   2  NUMBERS MULTIPLICATION OF WHICH
   WILL BE EQUAL TO  0.12500000


PRINT MINIMUM TRANSMISSION RATIO OF GROUP    1

   REQUIRED MINIMUM TRANSMISSION RATIOS

     0.24100000 = 0.241000 *

   REST OF MULTIPLICATION OF MINIMUM
   TRANSMISSION RATIOS MUST BE EQUAL TO         0.51867217
```

```
PRINT MINIMUM TRANSMISSION RATIO OF GROUP   2

   REQUIRED MINIMUM TRANSMISSION RATIOS

     0.12049997 = 0.241000 * 0.500000 *

   MINIMUM SPEED WILL BE =    108.45
   WITH  \3.60 % THEORETICAL ERROR

OPTION FOR MINIMUM TRANSMISSION RATIO  : CONT

   GEAR RATIO LESS THAN  0.2500  OR
   GEAR RATIO GREATER THAN  2.4000  IS NOT ACCEPTED
   YOU CAN NOT CONTINUE BEFORE ENTERING TRANSMISSION RATIOS WITHIN LIMITS

OPTION FOR MINIMUM TRANSMISSION RATIO  : POW

PRINT VALUES OF POWERS OF PROGRESSION RATIO
   TO OBTAIN MINIMUM TRANSMISSION RATIOS
   BY AN INCREASING ORDER FROM FIRST TO LAST TRANSMISSION
   YOU WILL PRINT   2  NUMBERS TOTAL OF WHICH IS EQUAL TO   6

   CALCULATED VALUES ARE   2 +  4 +

PRINT POWER OF PROGRESSION RATIO FOR GROUP  1

   REQUIRED POWERS OF PROGRESSION RATIO
   FOR MINIMUM TRANSMISSION RATIOS

 1.00 =  1.00 +

   REQUIRED MINIMUM TRANSMISSION RATIOS

     0.70921973 = 0.709220 *

   REST OF SUM OF POWERS OF
   PROGRESSION RATIO MUST BE EQUAL TO  5.00

PRINT POWER OF PROGRESSION RATIO FOR GROUP  2

   REQUIRED POWERS OF PROGRESSION RATIO
   FOR MINIMUM TRANSMISSION RATIOS

 6.00 =  1.00 +  5.00 +

   REQUIRED MINIMUM TRANSMISSION RATIOS

     0.12725912 = 0.709220 * 0.179434 *

   MINIMUM SPEED WILL BE =    114.53
   WITH   1.61 % THEORETICAL ERROR

OPTION FOR MINIMUM TRANSMISSION RATIO  : CONT

   GEAR RATIO LESS THAN  0.2500  OR
   GEAR RATIO GREATER THAN  2.4000  IS NOT ACCEPTED
   YOU CAN NOT CONTINUE BEFORE ENTERING TRANSMISSION RATIOS WITHIN LIMITS
```

OPTION FOR MINIMUM TRANSMISSION RATIO : CAL

REQUIRED MINIMUM TRANSMISSION RATIOS

0.12725816 = 0.502993 * 0.253002 *

MINIMUM SPEED WILL BE = 114.53
WITH 1.81 % THEORETICAL ERROR

OPTION FOR MINIMUM TRANSMISSION RATIO : CONT

FOUND TEETH NUMBERS FOR MINIMUM TRANSMISSION RATIOS

GRUP NUMB	REQUIRED RATIO	TEETH NUMBERS	FOUND RATIO	PERCENTAGE ERROR
1	0.502993	18 / 36	0.500000	\0.595047
2	0.253002	22 / 87	0.252874	\0.050784

FOR ALL OF THE OPTIONS (CONSIDERING THE ABOVE TEETH NUMBERS)

MINIMUM SPEED WILL BE 113.79
WITH \1.15 % ERROR WRT REQUIRED MINIMUM SPEED
WITH \0.65 % ERROR WRT REQUIRED TRANSMISSION RATIOS

PROPORTIONAL RADIAL DIMENSION OF GEARBOX WILL BE 163

THERE ARE 2 DIFFERENT SPEED DISTRIBUTION
THERE WILL BE 2 OPTIONS FOR EACH SPEED DISTRIBUTION
SAME ONES WILL BE ELIMINATED

OPTIONS FOR PRINTOUTS

ALL : TO GET ALL PRINTOUTS ONE BY ONE
REC : PRINTOUTS FOR ONLY RECOMMENDED OPTIONS

ANSWER FOR PRINTOUTS : ALL

OUTPUT RESULTS WILL BE STORED ON DISK

SPEED DISTRIBUTION FOR THE FOLLOWING OPTIONS

1. GROUP HAS 2 TRANSMISSIONS
2. GROUP HAS 3 TRANSMISSIONS

THERE WILL BE 2 DIFFERENT OPTIONS
FOR THIS SPEED DISTRIBUTION

************************ OPTION 1 ********************************

GRUP NUMB	SPED NUMB	POW.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
====	====	======	========	========	========	==========
1	2	0	0.502993	18 / 36	0.500000	\0.595047
		1	0.709220	22 / 31	0.709677	0.064497
2	3	0	0.253002	22 / 87	0.252874	\0.050784
		2	0.502993	36 / 72	0.500000	\0.595036
		4	1.000000	53 / 53	1.000000	0.000018

SHFT = 1

900.00			

SHFT = 2

450.00	638.71		

SHFT = 3

113.79	161.51	225.00	319.35
450.00	638.71		

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
=====	========	=======	=======	==============	==============
1	114.53	113.79	\0.65	0.40133799E\08	0.11426702E+09
2	161.49	161.51	0.01	0.33384489E\08	0.21151877E+09
3	227.70	225.00	\1.19	0.36492647E\08	0.18150314E+09
4	321.06	319.35	\0.53	0.26049103E\08	0.34697037E+09
5	452.69	450.00	\0.60	0.36029231E\08	0.27464704E+09
6	638.39	638.71	0.06	0.25115519E\08	0.53461504E+09

TOTAL ELASTICITY CONSTANT = 0.19720471E\07

TOTAL INERTIA CONSTANT = 0.16635210E+10

SUM OF LARGEST GEAR RATIOS = 1.709677

MAXIMUM GEAR RATIO OF OPTION = 1.000000

ORDER RANGE OF THIS OPTION = 1.254597

****

************************ OPTION 2 *******************************

GRUP NUMB	SPED NUMB	NUM.OF PRG.RT	REQUIRED RATIO	TEETH NUMBEERS	FOUND RATIO	PERCENTAGE ERROR
1	2	0	0.502993	18 / 36	0.500000	\0.595047
		3	1.409999	31 / 22	1.409090	\0.064396
2	3	0	0.253002	22 / 87	0.252874	\0.050784
		1	0.356733	29 / 81	0.358025	0.362166
		2	0.502993	36 / 72	0.500000	\0.595036

SHFT = 1

900.00

SHFT = 2

450.00 1268.13

SHFT = 3

113.79 161.11 225.00 320.69

454.04 634.09

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA -
1	114.53	113.79	\0.65	0.40133799E\08	0.11426792E+09
2	161.49	161.11	\0.23	0.37508450E\08	0.15639274E+09
3	227.70	225.00	\1.19	0.36492647E\08	0.18150314E+09
4	321.06	320.67	\0.12	0.56768137E\08	0.13137285E+10
5	452.67	454.04	0.30	0.56768137E\08	0.13137285E+10
6	638.30	634.09	\0.66	0.56768137E\08	0.13137285E+10

TOTAL ELASTICITY CONSTANT = 0.32142587E\07

TOTAL INERTIA CONSTANT = 0.36599196E+10

SUM OF LARGEST GEAR RATIOS = 1.909090

MAXIMUM GEAR RATIO OF OPTION = 1.409091

ERROR RANGE OF THIS OPTION = 1.487852

****

************************ OPTION 3 ********************************

GRUP NUMB	SPED NUMB	POW. OF PRG.RT	REQUIRED RATIO	TEETH NUMBERS	FOUND RATIO	PERCENTAGE ERROR
1	3	0	0.502993	18 / 36	0.500000	\0.595047
		1	0.709220	22 / 31	0.709677	0.064427
		2	1.000000	26 / 26	1.000000	0.000012
2	2	0	0.253002	22 / 87	0.252874	\0.050784
		3	0.709220	44 / 62	0.709677	0.064506

SHFT = 1

900.00

SHFT = 2

450.00 633.71 900.00

SHFT = 3

113.79 161.51 227.59 319.35

453.28 638.71

SPEED NO	REQUIRED SPEED	FOUND SPEED	% ERROR	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA
1	114.53	113.79	\0.65	0.40133799E\08	0.11426702E+09
2	161.49	161.51	0.01	0.33384489E\08	0.21151877E+09
3	227.70	227.59	\0.05	0.44449072E\08	0.39945370E+09
4	321.06	319.35	\0.53	0.36041046E\08	0.20969301E+09
5	454.69	453.28	0.13	0.25139322E\08	0.40376038E+09
6	639.30	638.71	0.06	0.28077014E\08	0.78115696E+09

TOTAL ELASTICITY CONSTANT = 0.20722369E\07

TOTAL INERTIA CONSTANT = 0.21198495E+10

SUM OF LARGEST GEAR RATIOS = 1.709677

MAXIMUM GEAR RATIO OF OPTION = 1.000000

ERROR RANGE OF THIS OPTION = 0.774834

****

```
************************   OPTION      4   ******************************

GRUP  SPED  POW.OF   REQUIRED     TEETH      FOUND      PERCENTAGE
NUMB  NUMB  PRG.RT    RATIO      NUMBERS     RATIO        ERROR
====  ====  ======   ========   ========   ========    ==========

  1     3     0      0.502993   18 / 36    0.500000    \0.595047
              2      1.000000   26 / 26    1.000000     0.000012
              4      1.988099   34 / 17    2.000000     0.595658

  2     2     0      0.253002   22 / 87    0.252874    \0.050784
              1      0.356733   29 / 81    0.358025     0.362166

 SHFT =  1
              900.00

 SHFT =  2
              450.00        900.00        1800.00

 SHFT =  3
              113.79        161.11        227.59        322.22
              455.17        644.44



SPEED   REQUIRED   FOUND                   PROPORTIONAL       PROPORTIONAL
 NO      SPEED     SPEED    % ERROR        ELASTICITY           INERTIA
=====   ========   =======  =======      ==============     =============

  1      114.53    113.79   \0.65      0.40133799E\06     0.11426702E+09
  2      161.49    161.11   \0.23      0.37508450E\06     0.15639274E+09
  3      227.70    227.59   \0.05      0.20106544E\07     0.22045140E+10
  4      321.06    322.22    0.36      0.33946779E\09     0.56795674E+09
  5      452.69    455.17    0.55      0.20106544E\07     0.22045140E+10
  6      635.10    644.44    0.96      0.20106544E\07     0.22045140E+10

   TOTAL ELASTICITY CONSTANT     = 0.60017271E\07

   TOTAL INERTIA CONSTANT        = 0.49730847E+10

   SUM OF LARGEST GEAR RATIOS    =    2.358025

   MAXIMUM GEAR RATIO OF OPTION  =    2.000000

   ERROR RANGE OF THIS OPTION    =    1.606652

**************************************************************************
```

OPTION	PROPORTIONAL ELASTICITY	PROPORTIONAL INERTIA	SUM OF MAX GEAR RATIOS	MAXIMUM GEAR RATIO	% ERROR RANGE
1	0.19720471E\07	0.16635210E+10	1.709677	1.000000	1.25
2	0.32142587E\07	0.36599196E+10	1.909090	1.409090	1.49
3	0.20722368E\07	0.21198475E+10	1.709677	1.000000	0.77
4	0.60317271E\07	0.49730847E+10	2.358025	2.000000	1.61

OPTION :STR

STORED INPUT VALUES

```
NUMBER OF SPEEDS         =     6
NUMBER OF SHAFTS         =     3
SHAFTS WITH 1 TRANS.     =     0
INPUT SPEED              =   900.00
MINIMUM SPEED            =   112.50
MAXIMUM SPEED            =   626.97
PROGRESSION RATIO        =     1.4100
```

STORED LIMITATIONS

```
MINIMUM TEETH NUMBER     =    18
TOTAL TEETH NUMBER       =   120
% ERROR ON GEAR RATIO    = \+ 1.00 %
% ERROR ON OUTPUT SPEED= \+ 2.00 %
MINIMUM TRANS. RATIO     =     0.25
MAXIMUM TRANS. RATIO     =     2.40
```

STORED MINIMUM TRANSMISSION RATIOS

0.12725818 = 1.000000 * 0.502993 * 0.253002 *

OPTION :X