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Assoc. Prof. Dr. Ali Rıza TEKİN
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Assoc. Prof. Dr. Melda ÇARPINLIOĞLU
Chairman of the Department

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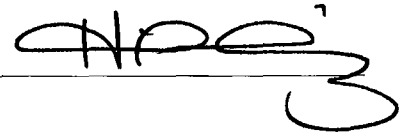
Asst. Prof. Dr. Cengiz DOĞAN
Supervisor

Examining Committee in Charge:

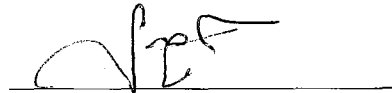
Prof. Dr. A. İhsan SÖNMEZ (Chairman)



Prof. Dr. İ. Hüseyin FİLİZ



Asst. Prof. Dr. Cengiz DOĞAN



Asst. Prof. Dr. Nihat YILDIRIM



Asst. Prof. Dr. Günseli GÖRÜR



ABSTRACT

COMPUTER AIDED DESIGN AND OPTIMIZATION OF GEAR REDUCTION UNIT

ÇETİNKAYA, Şükrü
M.S. in Mechanical Eng.
Supervisor: Asst. Prof. Dr. Cengiz DOĞAN
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In this thesis, a study on the design and optimisation of single and double stage gear reduction unit is presented . Theories on the design of gear pairs (spur, helical) are mentioned.

For the optimization process, linear searching method was used by taking the volume occupied by the pinion as objective function.

The iteration technique is used in determining gear parameters. A computer program was prepared to automate the procedure in FORTRAN 90 to accomplish the iteration processes, based on linear searching method and the assumptions made.

The program is tested with practical examples. Quite reasonable results (gear parameters) are obtained for the working condition of some of the commercially available gear reduction units of vehicles and engineering equipments.

Keywords: CAD, Gear Reduction Unit, Optimization of Gear Design

ÖZET

DİŞLİ REDÜKTÖRLERİNİN BİLGİSAYAR YARDIMIYLA TASARIMI ve OPTİMİZASYONU

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Bu tezde, tek ve çift kademeli redüktörlerdeki dişli çiftlerinin tasarımı ve optimizasyonu yapıldı. Dişli tasarımında kullanılan ana teorilerden bahsedildi.

Optimizasyon işlemi için, dişlinin kapladığı hacim objektif fonksiyon olarak alındı ve doğrusal tarama yöntemi kullanıldı.

Dişli parametrelerinin belirlenmesinde iterasyon tekniği kullanıldı. Varsayılan değerler üzerinde doğrusal araştırma yöntemi kullanıldı ve gerekli iterasyon işlemleri için FORTRAN 90 dilinde bir bilgisayar programı yazıldı.

Program, pratikte kullanılan örneklerle test edildi. Elimizde hazır bulunan ve aynı zamanda bazı araç ve mühendislik donanımları için kullanılan redüktörler için yapılan hesaplamalarda oldukça uygun dişli parametreleri elde edildi.

Anahtar Kelimeler: Bilgisayar Destekli Tasarım, Redüktörler, Optimum Dişli Tasarımı,

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NOMENCLATURE

<u>ITEM</u>	<u>DESCRIPTION</u>
Φ	Pressure Angle
F	Face width of gear
m	Module
m_n	Normal Module
ψ	Helix Angle
k_a	Surface Factor
k_b	Size Factor
k_c	Reliability Factor
k_d	Temperature Factor
k_e	Modifying Factor for Stress Concentration
k_f	Miscellaneous effects Factor
S_o	Endurance Limit of Mechanical Element (MPa)
S_o'	Endurance Limit of Rotating Test Specimen(MPa)
n	Factor of Safety
m_c	Contact Ratio
q_a	Arc of Approach
q_r	Arc of Recess
d	Pitch Diameter
P	Circular Pitch
K_o	Overload factor
K_m	Load distribution factor
V	Speed
n	Rotational Speed (Rev/min)
W_t	Transmitted Load
H	Transmitted Power (HP)
J	AGMA Geometry Factor for Bending

K_v	Velocity Factor
σ	Bending Stress
S_{ut}	Ultimate Tensile Strength
S_c	Surface Fatigue Strength
S_H	Hertzian strength
C_L	Life factor
C_H	Hardness ratio factor
C_T	Temperature factor
C_R	Reliability factor
W_{tp}	Permissible Transmitted Load
C_o	Overload Correction Factor
C_m	Load Distribution Factor
C_p	Elastic Coefficient
I	Geometry Factor for Surface Durability
P_n	Normal Circular Pitch
P_t	Transverse Circular Pitch
P_x	Axial Pitch
Φ_n	Normal Pressure Angle
Φ_t	Transverse Pressure Angle
m_N	Load Sharing Ratio
P_N	Normal Base Pitch
Z	Length of Line of Action
r_p	Pitch Radii of Pinion
r_G	Pitch Radii of Gear
r_{bp}	Base Circle Radii of Pinion
r_{bG}	Base Circle Radii of Gear
a	Addendum
GRU	Gear Reduction Unit
CD1	Center Distance for First Stage
CD2	Center Distance for Second Stage

CHAPTER 1

INTRODUCTION

In technology, gear reduction units are one of the most important tools that can be used to transmit power. In most of the machines, gearbox is used for obtaining various speed ranges, and for regulating torque requirement for the system.

In designing the gear trains, kinematic performance is an important factor, but it is equally important that the teeth are strong enough to withstand the applied torque. As torque is transmitted from one shaft to another, gear force is created along the pressure line. Gear teeth must be strong to resist this force. A satisfactory resistance of the tooth surface in meshing gears is also necessary.

Design of gears is usually based on three kinds of failures.

1. Static failure of the teeth of the gear due to bending stress.
2. Fatigue failure of the teeth of the gear due to bending stress.
3. Surface fatigue failure of the teeth due to contact or Hertzian stress.

Classical design of spur and helical gears are trial and error based. A preliminary design (module and pitch diameter) is assumed first and then, the strength calculations are performed. To prevent tooth breakage and pitting at the tooth surfaces of the gearset, wear and breakage strength must satisfy the safety requirements. If these conditions are not satisfied, then all the assumptions are changed and all the procedure is repeated until a satisfactory solution is reached. However, in addition to the time lost, optimum solutions may not be achieved. For

more complicated gear systems the number of constraints increase, and the design procedure becomes complex and needs more iterations.

As a result of these problems, a method of computer aided design is performed for spur and helical gear systems, by which quick preliminary solutions and optimised solutions can be obtained.

For the design of spur and helical gears, classical rating formulas have been used after arranging in a consecutive form to determine face widths and modules. Bending fatigue strength and surface fatigue strength are the design criteria for gears.

Inertia is one of the main parameter that should be considered in the design of gear reduction unit. So, weights (dimensions) of the gears are the most important parameters that effect the inertia of the system. In designing gear reduction unit, different pinion and gear sizes can satisfy all the requirements of strength, wear, speed ratio etc., but determining the one with smallest volume or inertia is the main objective.

Considering the problems encountered in the previous gear design works and the opportunity of applying optimisation methods, it is aimed to develop a computer aided design technique, so that optimum design of gears with accurate and rapid calculations would be possible in a short time.

In this study, a personal computer was used and a program was written in Fortran 90 programming language for the design and optimisation processes. In the first place, attempts are made to set up design principles of gear reduction unit and to automate the stages that indicates the gear dimensions used in a reduction unit. The usage of computer helped us to perform design operations in short time, and to indicate the optimum conditions for the application of the gears.

The optimisation criteria for the program is to design the gears with minimum volume. This is done by linear searching method. For a double stage gear reduction unit, the program optimises the solutions with respect to the different velocity ratios,

generated by the program, for both the first and the second stages, while keeping the overall velocity ratio constant.

The method presented here pinpoints the gear sizes and ratios that will permit the lightest possible design while still meeting the velocity ratio, strength, and surface durability requirements of the application. The method is applicable to both spur and helical gears.



CHAPTER 2

LITERATURE SURVEY

2.1 INTRODUCTION

This chapter presents a brief survey of the most relevant literature related to the work reported in this thesis. In section 2.2, studies on the Computer Aided Design of gears are reviewed. Studies about optimization of gears are presented in section 2.3.

2.2 COMPUTER AIDED DESIGN OF GEARS

Chironis [1] studied the determination of gear teeth number.

Orthwein[2] and Reddy[3] proposed methods to find gear teeth number for specified error range and for a given centre distance.

Strength of a gear tooth was first calculated to a close degree of accuracy by Willfred LEWIS in 1892. He conceived the idea of inscribing a parabola of uniform strength inside a gear tooth. He observed that when a parabola is made into a cantilever beam, the stress is constant along the surface of the parabola [23].

One of the most commonly used text by the gear designers is 'Gear Hand Book', published by American Gear Manufacturers Association[7]. Besides the

condensed information on many types of gears that are now in use, specific spur and helical gear calculation procedures are also given. It is summarised as; [20]

a) The procedure starts with the given application requirements that include the speed ratio, input speed, and kind of duty to be performed. The duty may be one of power or motion transmission.

b) Based on application requirements, the standard pressure angle is selected from the given table.

c) The approximate number of pinion teeth is picked from the table which is prepared for general applications. This value is checked for undercut limitations with the use of tables. If the design is, for power gearing, number of pinion teeth is checked for data on balancing strength and wear capacity, again with the use of tables.

d) Having approximate number of pinion teeth, appropriate centre distance and face width are determined with the basic equations.

e) With the number of pinion teeth and centre distance, pitch is determined. Pinion teeth numbers, centre distance and ratio are readjusted to agree with pitch chosen. If the readjusted values do not satisfy the application requirements, the procedure is repeated.

f) Finally the basic tooth proportions of gear are determined. A similar approach is also used by Darle W. Dudley[8].

Virgil Moring Faires, Anthony Esposito, Paul H. Black [9, 10, 11], use modified Lewis Equation, to calculate bending strength of gear teeth. Esposito modifies the equation by dynamic correction factor which is a function of pitch line velocity only. On the other hand Faires and Black modify by strength reduction factor, within the range of 1.2 to 1.7, depends on operating conditions. Dynamic load and limiting wear load are calculated by using Buckingham's equations by all of them. Faires uses simpler equations when the service is intermittent and wear is not a factor,

for calculating dynamic load, which are functions of pitch line velocity. The breakage and wear loads are compared with the dynamic load to see if they are equal or greater than the dynamic load. If it is not satisfied then the predetermined independent geometrical or material parameters are changed and all the calculations are repeated. [20]

R. G. Mitchiner and H. H. Mabie [14] investigated tooth stresses. They presented a simple and direct approach to the problem of definition of the root profile for standard and nonstandard external spur gear teeth. They gave equations and developed a computer program to compute the Lewis Form Factor Y, and the AGMA Geometry factor J. They located the point of interest on the root profile to calculate stresses developed. This work is also reported by Shigley in Mechanical Engineering Design book [15].

Yelle[16] developed a set of equations and programmed them in BASIC to calculate the Lewis Form Factor Y, by tracing the procedure stated in Ref.[17]. He further calculated the tooth deformation, the load transferring points and the load sharing for a large variety of gear pair geometries and material combinations.[23]

Dudley [18] reported the equations for stress concentration factor developed by Dolan and Broghamer in 1942. He also reported the work of the research committee of the ASME, headed by Earle Buckingham [19], on the dynamic gear tooth load. Shigley mentioned about Carly G. Barth who first introduced dynamic factor in nineteenth century.

Another design procedure for spur and helical gears, given by J. E. Shigley[12] is as follows:[20]

a) Bending stress is calculated by Lewis equation which is modified by dynamic and geometric factors.

b) Endurance limit strength of the gear tooth, due to bending, is then calculated by taking surface, size, reliability, temperature, stress concentration and the

other miscellaneous effects into account. All the above factors are determined from AGMA standard[17].

c) The desired factor of safety is calculated with the given equation which includes overload and load distribution factors. On the other hand the real factor of safety is calculated by using endurance and bending strength of the tooth. If real factor of safety is equal or greater than the desired one, then it is concluded that gears will have an infinite life. Otherwise, design parameters are changed accordingly until a satisfactory result is obtained.

d) The surface durability is calculated with a given equation which involves elastic, velocity and geometry factors, given by tables. The surface endurance limit is modified by using life, satisfactory life, the dynamic surface stress is checked whether it is within the surface endurance limit of the gear material or not.

Karlı and Filiz[4] and Karlı[5] reviewed analytical method of obtaining kinematical arrangement diagram and prepared a computer aided design package to obtain all possible kinematical arrangements of gearbox within the 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.

Dölger[23] reviewed general types of gearboxes and theories of the design of the main gearbox elements. A design package in which gear and shaft designs based on both strength and deflection mode are fully automated. For this purpose a highly user oriented computer program is written in Fortran 77. By providing standard data for the design, he obtained modules and face widths of gears. He obtained quite conservative results for gearboxes already used in existing machine tools.

Bilgin Kaftanoğlu and Nuri Çarkoğlu [29] developed a computer program to design spur and helical gears. The AGMA standard 218.01 was used to design the gears. The modules selected for the design were the standard or recommended modules.

Bilgin Kaftanoğlu and Nuri Çarkoğlu[29] prepared a different computer program to design several mechanical power transmissions elements, such as power shafts, brake and clutches, spur and helical gears.

2.3. OPTIMUM DESIGN OF GEARS

An analytical method of optimum design of gears is developed by Ray.C. Johnson[13]. Shortly with this method a designer is able to select optimum material and is able to determine the optimum geometry for the gears.[20]

Turkmen [20] prepared a design package which optimise the gear transmissions by achieving the minimum permissible centre distance. He tested his program for gear transmission systems with single or double reduction, namely, gearbox of FIAT model 480 tractor and gearbox of Murat 131 Automobile. He got quite compatible results with the actual values.

Inertia of gear trains is studied by Wright [21] and Selfridge[22]. They presented equations to calculate equivalent inertia of a compound gear train.

The design of a pair of internal spur and helical gears was treated with the objective of minimising the centre distance or gear volume for a given specification by B.S. Tong and Walton [24]. The designs obtained satisfy all the necessary conditions for kinematics limits on contact ratio, interference and gear ratio tolerance and met strength and wear criteria according to the design standard selected.

A practical procedure for designing optimum helical gears was shown by Rodrigo Lopez Sansalvador and Juan Carlos Jaurequi C. [25]. The optimization procedure was adapted to technical limitations, and it was focused on real world cases. Limiting parameters and restrictions were defined. They described a graphical solution. According to their study, a design procedure has four steps:

1. Recognition of a need,
2. Statement of the problem,
3. Creation of alternative solutions,
4. Selection of alternatives.

G. Madhusudan and C.R. Vijayapimha [26] prepared a computer program capable of designing a required type of gear under a specified set of working conditions. This study was oriented towards the training and education aspects of the gear design process.

M. Savege, J.J. Coy and D.P. Townsend [27] treated the design of a standard gear mesh with the objective of minimising the gear size for a given ratio, pinion torque, and allowable tooth strength. Scoring, pitting fatigue, bending fatigue, and the kinematic limits of contact ratio and interference were considered. A design space was defined in terms of the number of teeth on the pinion and the diametral pitch. This space was then combined with the objective function of minimum centre distance to obtain an optimal design region. This region defines the number pinion teeth for the most compact design.

A computer program is developed by D. Frayne [6]. The developed basic program, for spur and helical gears, simplifies the design procedure, and is essentially a minimum weight design. It considers both the gear structure and gear life. The main advantage of the program is the simple language used for typing the input data into the computer and having all executed data easily readable[20].

X.Li., G.R. Symmons and G. Cockerham [28] presented a method for minimising the centre distance of a helical gear set of prescribed power capacity using a fixed ratio of facewidth to pcd and formulae contained within AGMA procedures. Although power capacity is dependent upon many factors, the method has used module, numbers of teeth, and helix angle as the significant variables. The procedure was based upon determining limiting values of AGMA geometry factors I and J which were derived in order to establish upper and lower boundaries for the search interval of centre distance.

S. Prayoonrat and D. Walton [30] prepared an algorithm to design and optimise multispindle gear trains of the nonspeed-change type. The designer may choose to optimise gear trains on the basis of minimum overall centre distance, minimum overall size, minimum gear volume, or other desirable criteria, such as maximum contact or overlap ratios. The method was based on a two-stage optimization procedure. A direct search method was used to remove unacceptable solutions and to locate quickly the region where the optimum design lies.

B.S. Tong and D. Walton [31] described an interactive program to design internal gear pairs. From a specification, the program first performs a kinematic analysis to determine tooth numbers and to satisfy centre distance requirements, and then proceeds to calculate the facewidth.

An efficient method for analysing Multispeed gear trains was presented and used by G. S. Bush, M. O. M. Osman and S. Sankar[32].

Douglas Hughson[33] developed a new computerised gear system which tailors the design of parallel axis gears. It minimises surface stress for tractor and truck gears and maximises contact ratio for automobile gears. They equalised the bending fatigue time life of the pinion to gear on demand and balances the resistance to bending fatigue with the resistance to surface damage.

CHAPTER 3

GEAR DESIGN

3.1 INTRODUCTION

In this chapter, general design procedure of the gears is described. The design procedure is then applied to gear transmission systems with single reduction by taking all probable applications into account.

For the sake of simplicity and conformity, the procedures are given in metric system. The developed computer program is also able to perform the calculations in British System by changing formulas and tables, whenever desired.

Section 3.2 is devoted to gear design in which the spur gear geometry and basic principles of the gear design are discussed and formulation is presented.

Section 3.3 is arranged for indicating the rules for designing of helical gears and 3.4 is about the material that can be used for the gears production.

3.2 SPUR GEARS

The spur gears have teeth on the outside of a cylinder and the teeth are parallel to the axis of the cylinder. This simple type of gear is the most common type. Its volume of usage is the largest of all types.

Lower pressure angles have the advantage of smoother and quieter tooth action because of the large profile contact ratio. In addition, lower loads are imposed

on the support bearings because of a decreased radial load component; however, the tangential load component remains unchanged with pressure angle. The problem of undercutting associated with small numbers of pinion teeth is more severe with the lower pressure angle. Lower pressure angle gears also have lower bending strength and surface durability ratings and operate with higher sliding velocities than their higher pressure angle counterparts.

Higher pressure angles have the advantage of better load-carrying capacity, with respect to both strength and durability, and lower sliding velocities (thus better scoring and wear performance characteristics). In some cases, very high pressure angles, 28, 30, and, in a few cases, as high as 45°, are employed in some special slow-speed gears for very high load capacity where noise is not the predominant consideration. [43]

3.2.1 Spur Gear Geometry

In spur gears, the teeth are straight and parallel to the axis of rotation. The terminology of teeth of spur gears is shown in figure 3.1.

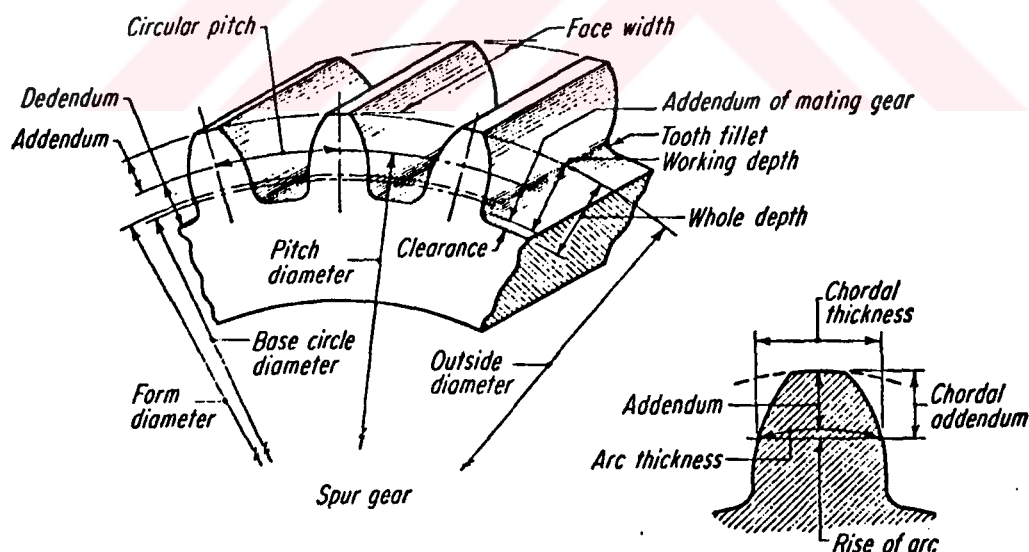


Figure 3.1. Nomenclature of Spur Gear Teeth[41]

Theoretical circle that all calculations are usually based on is named as pitch circle. The smaller of two mating gears is called the pinion. The larger is called the gear.

The circular pitch p is the arc length on the pitch circles between successive tooth profiles, and is the same for both gears and the basic rack. Thus the circular pitch is equal to the sum of the tooth thickness and the width of space.

The ratio of the pitch diameter to the number of teeth is named as module. For properly mesh, both the pinion and gear must have exactly the same module. Each module requires a separate cutting tool. For this reason, only a finite number of modules have been standardised and those standard modules are listed in table 3.1.

Table 3.1. Standard Modules[40]

Preferred	1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40, 50
Next choice	1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18, 22, 28, 36, 45

The circle that is tangent to the addendum circle of the mating gear is named as clearance circle. The clearance c is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear. The circular thickness of the teeth is measured along the pitch circle; the sum of the circular thickness of a pair equals the circular pitch minus the backlash. The distance that the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circles is named as backlash. Backlash is the main drawback for the gear reduction unit. It cause noise and vibrations. For smooth power transmission, it should be controlled at reasonable level.

The relation of module with pitch diameter can be written as,

$$m = \frac{d}{N} \quad (3.1)$$

where m = module, mm

d = pitch diameter, mm

N = number of teeth

$$p = \frac{\pi d}{N} = \pi m \quad (3.2)$$

where p is the circular pitch, mm

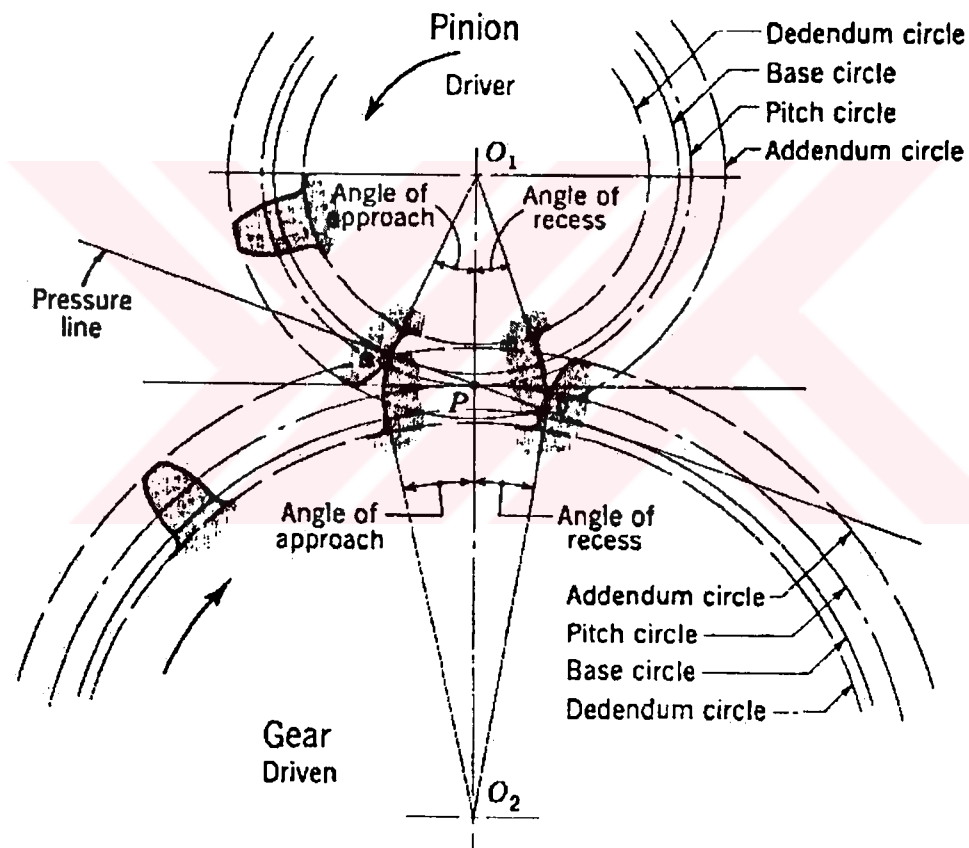


Figure 3.2. Tooth Action

3.2.2 Tooth Action

In order to explain the tooth action, first we must examine figure 3.3. The pinion with centre at O_1 turns counter clockwise. When the flank of the driver comes into contact with the tip of the driven tooth on generating line (point a), the initial contact will take place. During meshing of the teeth, the point of contact will slide up the side of the pinion tooth, so that, the tip of the driver will be in contact just before contact ends (point b). The final point of contact is the point where the addendum circle of the driver crosses the pressure line.

When the gears are made accurate enough for the teeth to share load, the tip load condition, which the contact is at the tip of gear tooth, is not the most critical. In nearly all gear designs, the second pair of teeth is put into contact (contact ratio is greater than 1) when one pair has reached the tip load condition. Therefore the worst load condition occurs when a single pair of teeth carry full load and the contact has rolled to a point at which a second pair of teeth is ready to come into contact [23].

3.2.3 Contact Ratio

Due to nature of the tooth action, for the teeth are in contact, it is suitable to define the contact ratio m_c as

$$m_c = \frac{q_t}{p} \quad (3.3)$$

where q_t is arc of action.

$$q_t = q_a + q_r \quad (3.4)$$

where q_a is the arc of approach, q_r is the arc of recess, as shown in figure 3.3.

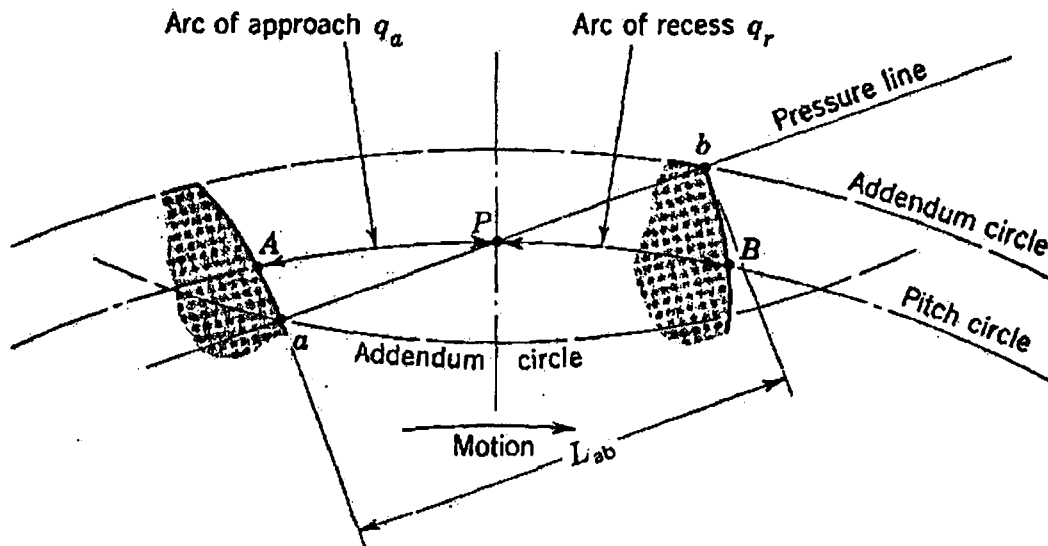


Figure 3.3 Definition of Contact Ratio

As specified in Ref. [43]; gears should not generally be designed having contact ratios less than about 1.20, because inaccuracies in mounting might reduce the contact ratio even more, increasing the possibility of impact between the teeth as well as an increase in the noise level.

3.2.4. Force Analysis

A pinion mounted on shaft a and driving a gear on shaft b are illustrated in figure 3.4a. Pinion is rotating clockwise at n_2 speed and gear at n_3 . The reactions between the gear teeth occur along the pressure line. The pinion has been separated from the gear and from the shaft in figure 3.4b, and their effects replaced by forces. Also, forces exerted by each gear are presented in figure 3.4b. F_{a2} and T_{a2} are the force and torque, respectively, exerted by shaft a against pinion 2. F_{32} is the force exerted by gear 3 against the pinion.

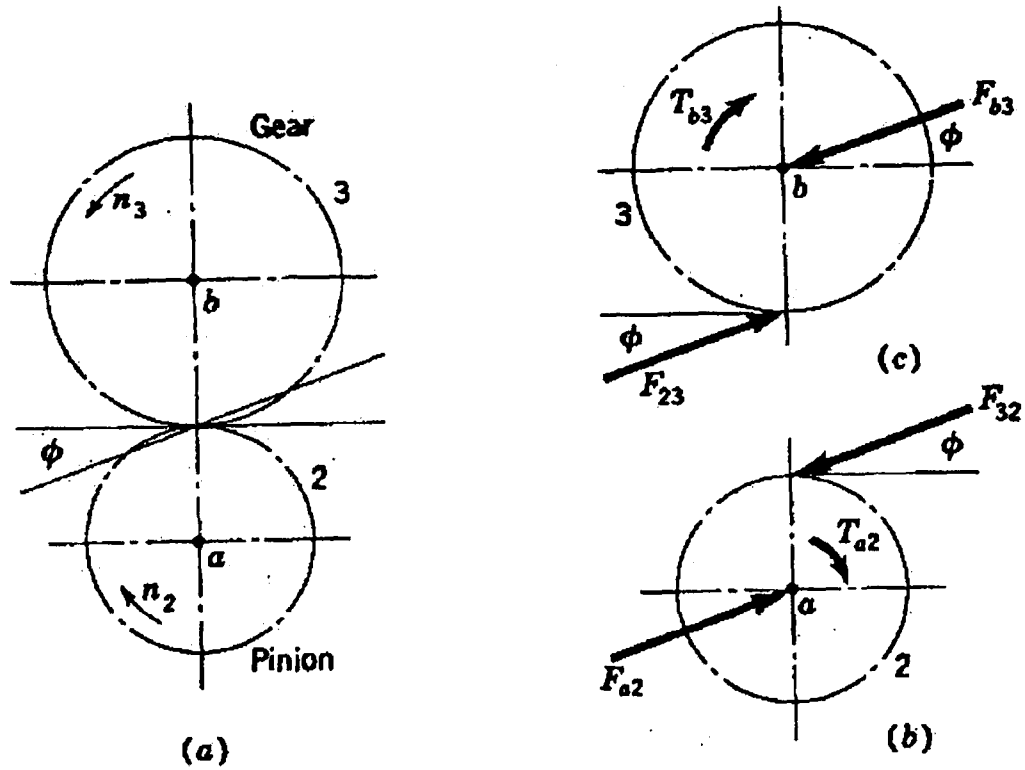


Figure 3.4. Free body diagrams of the forces acting upon two gears in a simple gear train[40]

The transmitted load is the main component. The radial component F_{32}^r does not transmit power. Power is transmitted by tangential component F_{32}^t and generally it is named as W_t .

The relation between the applied torque and the transmitted load can be written as

$$T = \frac{d}{2} W_t \quad (3.5)$$

where, $T = T_{a2}$ and $d = d_2$

The velocity in pitch line is

$$V = \frac{\pi d n}{60} \quad (3.6)$$

where V = pitch line velocity, m/s

d = pitch diameter, m

n = speed, rev/min

With these relations, the tangential load W_t can be obtained from the equation

$$H = W_t V \quad (3.7)$$

Where H is the transmitted power in watts.

3.2.5 Gear Design for Bending Strength

In this section it is tried to obtain the relations for the bending stresses in a loaded tooth. Wilfred Lewis first presented a formula for computing the bending stress in gear teeth. The formula was announced in 1892.

In Lewis analysis, the load sharing ratio and the fatigue stress concentration factor were not taken into account. A factor (J) called geometry factor has been established by AGMA. This factor is a modified form of Lewis form factor. Values of geometry factor J are given in tables A.1 to A.4. in the appendix for some of 20° and 25° standard pressure angles. All of tabular material in this section is taken from Mitchiner and Mabie paper [14].

When a pair of gears is driven at moderate or high speeds, dynamic effect is presented, and this factor is presented as K_v in bending stress expression. This dynamic effect is usually dependent on the pitch line velocity, material type, and the way of gear production.

First, the velocity factor (dynamic factor) was expressed in the nineteenth century by Carl G., and Barth by the equation

$$K_v = \frac{3}{3 + V} \quad (3.8)$$

for cast iron gears with cast teeth. This equation is modified to

$$K_v = \frac{6}{6 + V} \quad (3.9)$$

for cut or milled teeth or for gears not carefully generated.

For spur gears that are finished by hobbing or shaping, AGMA recommends the formula

$$K_v = \frac{50}{50 + \sqrt{200V}} \quad (3.10)$$

If the gears have high precision shaved or ground teeth, the AGMA dynamic factor is

$$K_v = \sqrt{\frac{78}{78 + \sqrt{200V}}} \quad (3.11)$$

In all of these equations V is the pitch line velocity in meter per minute. In this study, equation (3.11) was used for computing the dynamic factor in the computer program by the author.

Based on the above parameters, bending stress can be written as

$$\sigma = \frac{W_t}{F J m K_v} \quad (3.12)$$

Where W_t is the transmitted load, F is the face width and m is the module.

3.2.6. Fatigue Strength

The endurance limit of a gear tooth can be determined from

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \quad (3.13)$$

where, S_e = Endurance limit of gear tooth

S'_e = Endurance limit of rotating beam specimen.

k_a = Surface factor

k_b = Size factor

k_c = Reliability factor

k_d = Temperature factor

k_e = Modifying factor for stress concentration

k_f = Miscellaneous effect factor.

The endurance limit of rotating beam specimen is

$$\begin{aligned} S'_e &= 0.5 S_{ut} \quad \text{If} \quad S_{ut} < 1400 \text{ MPa} \\ S'_e &= 700 \text{ MPa} \quad \text{If} \quad S_{ut} > 1400 \text{ MPa} \end{aligned} \quad (3.14)$$

The surface factor (k_a) should be taken from figure 3.5. Always, it should correspond to a machined finish curve, even when the flank of gear tooth is ground or shaved. The reason for this is that the bottom surface is usually not ground, left as the original machined finish.

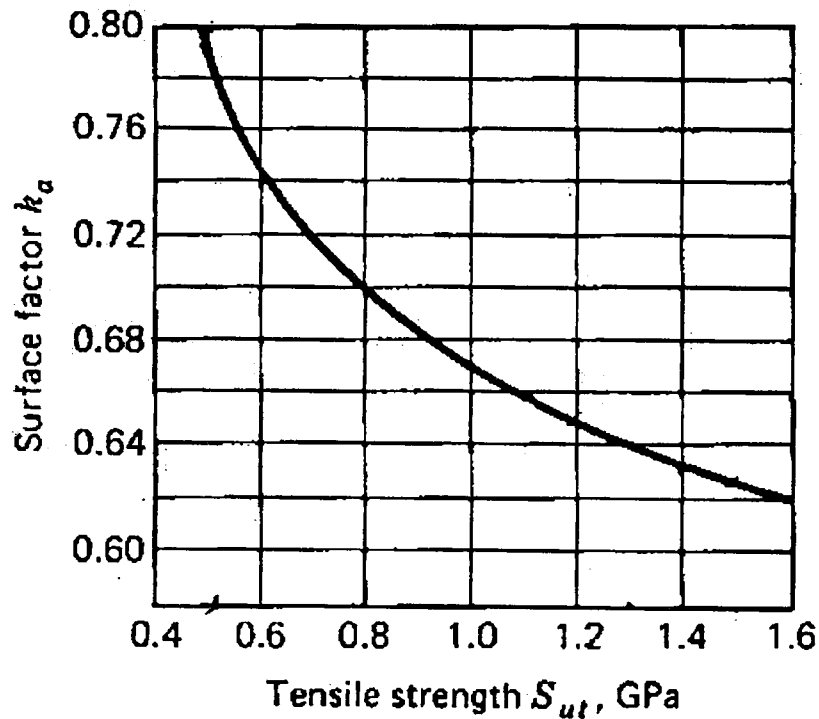


Figure 3.5 Surface Finish Factors k_a for Cut, Shaved, and Ground Gear Teeth

The size factors for gears are listed in Table A5 in appendix.

The necessary reliability factors are listed in Table 3.2.

Table 3.2 Reliability Factors

Reliability R	0.50	0.90	0.95	0.99	0.999	0.9999
Factor k_e	1.000	0.897	0.868	0.814	0.753	0.702

The properties of a material may be greatly changed by the temperature. For example, high temperature mobilise dislocations and reduce the fatigue resistance in many materials. For this reason, as specified in Ref. [23], laboratory testing is recommended when a new and untried materials are under considerations. These observations may be approximated by some relations for the engineers as stated below:

$$\begin{aligned}
k_d &= 1 & \text{If } T < 450 \text{ C}^\circ \\
k_d &= 1 - 0.0058(T - 450) & \text{If } 450 < T < 550 \text{ C}^\circ \\
k_d &= 0.4 & \text{If } T > 550 \text{ C}^\circ
\end{aligned} \tag{3.15}$$

The fatigue stress concentration factor has been incorporated into the geometry factor J. Hence, it is accounted for the gears as 1.00.

Miscellaneous effect factor (k_f) is taken from figure 3.6. Use $k_f = 1.33$ for values of S_{ut} less than 1.4 GPa.

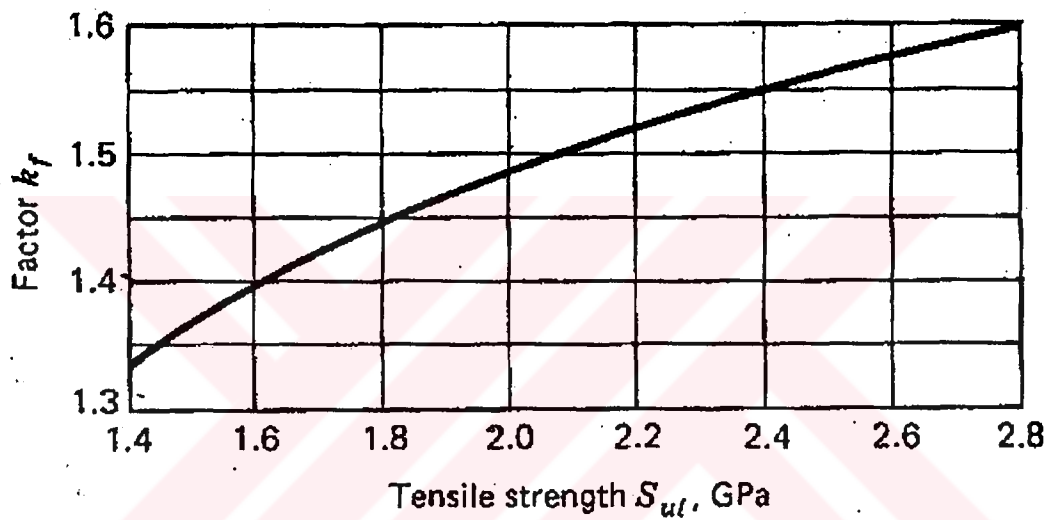


Figure 3.6 Miscellaneous Effects Factors for One Way Bending of Gear Teeth.

For the gear tooth, the factor of safety is calculated by the formula

$$n_G = K_o \cdot K_m \cdot n \tag{3.16}$$

where n_G : Factor of safety for gears

K_o : Overload factor

K_m : Load distribution factor

n : Usual factor of safety

The AGMA recommendation is to use $n \geq 2$ to guard against fatigue failure.

Overload factor makes allowances for roughness or smoothness of operation and, is based on service factors. The values of overload factor (K_o) are presented in table 3.3 according to the AGMA recommendation.

Factor K_m is an AGMA load distribution factor that accounts for the possibility that the tooth force may not be uniformly distributed across the full face width. Factors K_m are tabulated in Table A6 in appendix.

Table 3.3 Overload Correction Factor K_o

Source of power	Driven Machinery		
	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

3.2.7. Gear Design for Surface Durability

It should be appreciated in addition that fracture of gear teeth is not the only mode of failure to be considered but that wear of the teeth is also a criterion; in fact, more gears fail by wearing of the teeth than by fracture .

The durability of a gear set is usually evaluated by calculating the maximum surface contact stress on the gear tooth and then comparing this value with allowable values established by test or experience for a specific material or material combination. [42]

At high contact stresses, due to many repetitions, surface of gear tooth wears and this is known as surface fatigue failure. In general, this type of failure is named as pitting.

Frequently gears will wear to the point where they begin to run roughly. Then the increased in dynamic load plus the stress concentration effects of the tooth surface cause the teeth to fail by breakage.

The gear tooth surfaces move across each other with a combination of rolling and sliding motion. The sliding motion plus the coefficient of friction tends to cause additional surface stresses.

A bit of metal on the surface of a gear tooth goes through a cycle of compression and tension each time a mating gear tooth passes over it. If the tooth is loaded heavily enough, there will usually be evidence of both surface cracks and plastic flow on the contacting surface. There may also be a rupturing of the metal as a result of subsurface shear stresses.

H. Hertz developed a mathematical theory in the late nineteenth century, for the surface stresses and deformations produced when two curved bodies are passed together. Cylinder on cylinder with their axes parallel, the hertzian stress in a gear tooth is given by

$$\sigma = -C_p \sqrt{\frac{W}{F.I.C_v.d_p}} \quad (3.17)$$

Where C_p : Elastic coefficient
 I : Geometry factor
 d_p : Pinion diameter
 C_v : Velocity factor ($C_v = K_v$)

If the pinion and gear are produced from different materials, elastic coefficient can be determined by equation (3.17) and also can be obtained from Table A7 in appendix.

$$C_p = \sqrt{\frac{1}{\pi \left(\frac{1 - \nu_p^2}{E_p} + \frac{1 - \nu_G^2}{E_G} \right)}} \quad (3.18)$$

Where E is a young modules of elasticity and ν is a Poisson's ratio.

The geometry factor for external spur gears now, can be defined as

$$I = \frac{\cos\Phi \sin\Phi}{2} \frac{m_G}{m_G + 1} \quad (3.19)$$

Where Φ is a pressure angle and m_G is load sharing ratio (d_G/d_p). In equation (3.17), minus sign shows that Hertzian stress is always compressive.

The surface fatigue strength for steel is [23]

$$S_c = 2.76 \text{ HB} - 70 \text{ (MPa)} \quad (3.20)$$

Where HB is the Brinel Hardness of the softer of the two contacting surfaces.

As AGMA recommends, the surface fatigue strength is to be modified as

$$S_H = \frac{C_L \cdot C_T}{C_H \cdot C_R} S_c \quad (3.21)$$

Where S_H : Corrected fatigue strength, or Hertzian strength

C_L : Life factor (table 3.4)

C_H : Hardness ratio factor; for spur gears use 1.0

C_T : Temperature factor; for temperatures less than 120 C° use 1.0

C_R : Reliability factor listed in table 3.4 as recommended by the AGMA

Table 3.4 Life and Reliability Modification Factors [40]

Cycles of life	Life factor	Reliability R	Reliability factor
10000	1.5	Up to 0.99	0.80
100000	1.3	0.99 to 0.999	1.00
1000000	1.1	0.999 up	1.25 up
100000000 up	1.0		

The life modification factor C_L is used to increase the strength when the gear is to be used for short periods of time.

The permissible transmitted load W_{tp} is designated as

$$W_{tp} = n_G W_t \quad (3.22)$$

From equation (3.16)

$$n_G = C_o \cdot C_m \cdot n \quad (3.23)$$

As a design equation, corrected surface fatigue strength can be written as

$$S_H = C_p \sqrt{\frac{W_{tp}}{F \cdot I \cdot C_v \cdot d_p}} \quad (3.24)$$

3.3. HELICAL GEAR

In the case of helical gears, the teeth are not parallel to the axis of rotation. In helical gear design it is common practice to refer to the normal plane as well as the diametral plane. The normal plane is perpendicular to a tooth element on the pitch cylinder surface at the pitch point.

When gear teeth are cut on a spiral that wraps around a cylinder, they are designated as helical. Helical teeth enter the meshing zone progressively and, therefore, have a smoother action than spur gear teeth and tend to be quieter. In addition, the load transmitted may be somewhat larger, or the life of the gears may be greater for the same loading, than with an equivalent pair of spur gears. Conversely, in some cases smaller size helical gears (compared with spur gear) may be used to transmit the same loading. Helical gears produce an end thrust along the axis of the shafts in addition to separating and tangential (driving) loads of spur gears. Where suitable means can be provided to take this thrust, such as thrust collars or thruot or tapered roller bearings, it is no great disadvantage. The efficiency of a helical gear set, which is dependent on the total normal tooth load, will usually be slightly lower than for an equivalent spur gear set.

In order to provide significant improvement in noise over spur gears, the face overlap must be at least unity. That is, one end of the gear face must be advanced at least one circular pitch from the other. If the face overlap (also called face contact ratio) is less than unity, the helical gear, for most analytical purposes (that is, stress calculations) is treated as if it were a spur gear with no advantage taken of the fact that it is helical.

Helix angles from only a few degrees up to about 45° are practical. As the helix angle increases from zero, in general, the noise level is reduced and the load capacity is increased. At angles much above 15 to 20° , however, the tooth bending capacity is generally begins to drop off. This is due to the fact that the transverse tooth thickness decreases rapidly [43].

3.3.1. Kinematics of Parallel Helical gears

Helical gears are used to transmit motion between parallel shafts. Helical gears are used when both high speed and high horsepower are required. The helix angle is the same on each gear, but one gear must have a right hand helix and the other has a left hand helix. As shown in figure.3.7 the shape of the tooth is an involute helicoid.

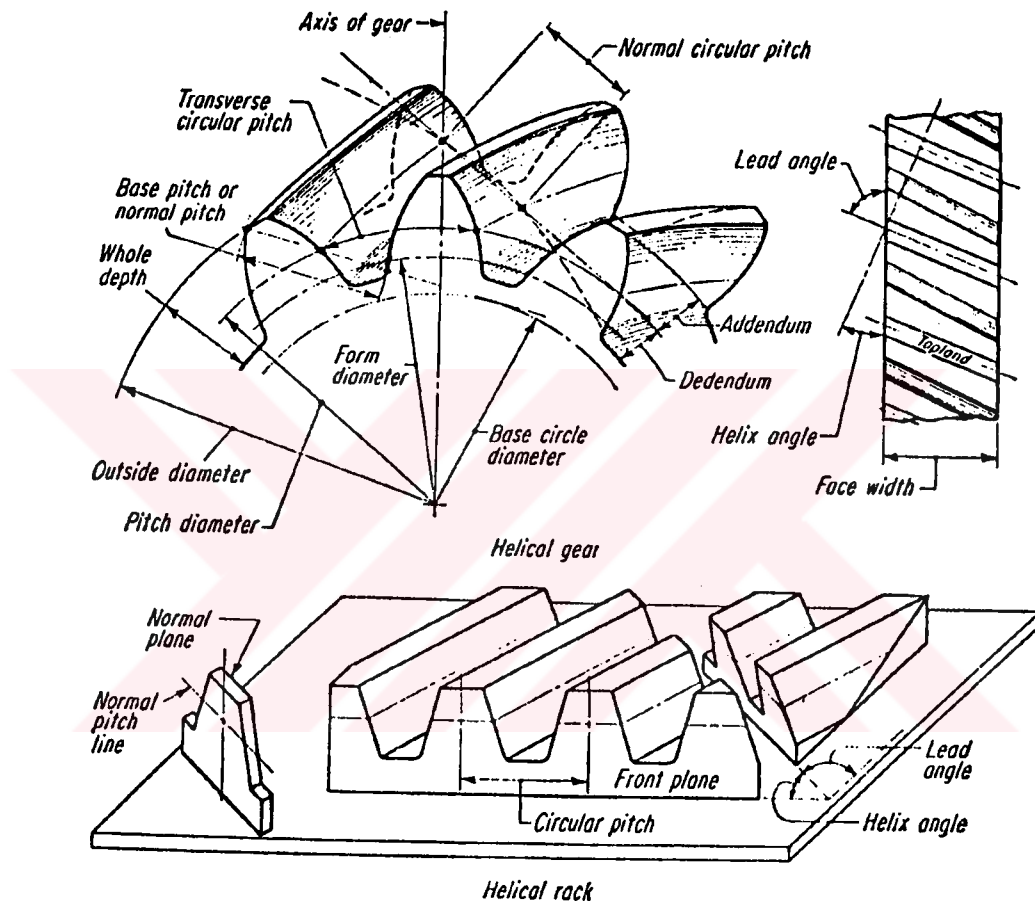


Figure 3.7. Helical Gear and Rack Terminology [41]

The circular pitch referred to the normal plane P_n is given by the following equation

$$P_n = P_t \cdot \cos \psi \quad (3.25)$$

Where P_n : Normal circular pitch

P_t : Transverse circular pitch

Ψ : Helix angle of the gear set at the pitch surface.

The relation between transverse circular pitch and axial pitch can be found by the expression;

$$P_x = \frac{P_t}{\tan \Psi} \quad (3.26)$$

The normal module m_n is

$$m_n = m \cdot \cos \Psi \quad (3.27)$$

Because of the angularity of the teeth, the pressure angle Φ_n in the normal direction is different from the pressure angle Φ_t in the direction of rotation. These angles are related by the equation;

$$\cos \Psi = \frac{\tan \Phi_n}{\tan \Phi_t} \quad (3.28)$$

As a general guide, tooth proportions should be based on a normal pressure angle of 20° . Many authorities recommend that the face width of helical gears be at least twice the axial pitch to obtain helical gear reduction.

$$F = 2P_x \quad (3.29)$$

3.3.2 Force Analysis

From the geometry present in figure 3.7, three components of total tooth force W are:

$$\begin{aligned}W_r &= W \cdot \sin\Phi_n \\W_t &= W \cdot \cos\Phi_n \cdot \cos\psi \\W_a &= W \cdot \cos\Phi_n \cdot \sin\psi\end{aligned}\tag{3.30}$$

Where W = total force

W_r = radial component

W_t = tangential component (transmitted load)

W_a = axial component (thrust load)



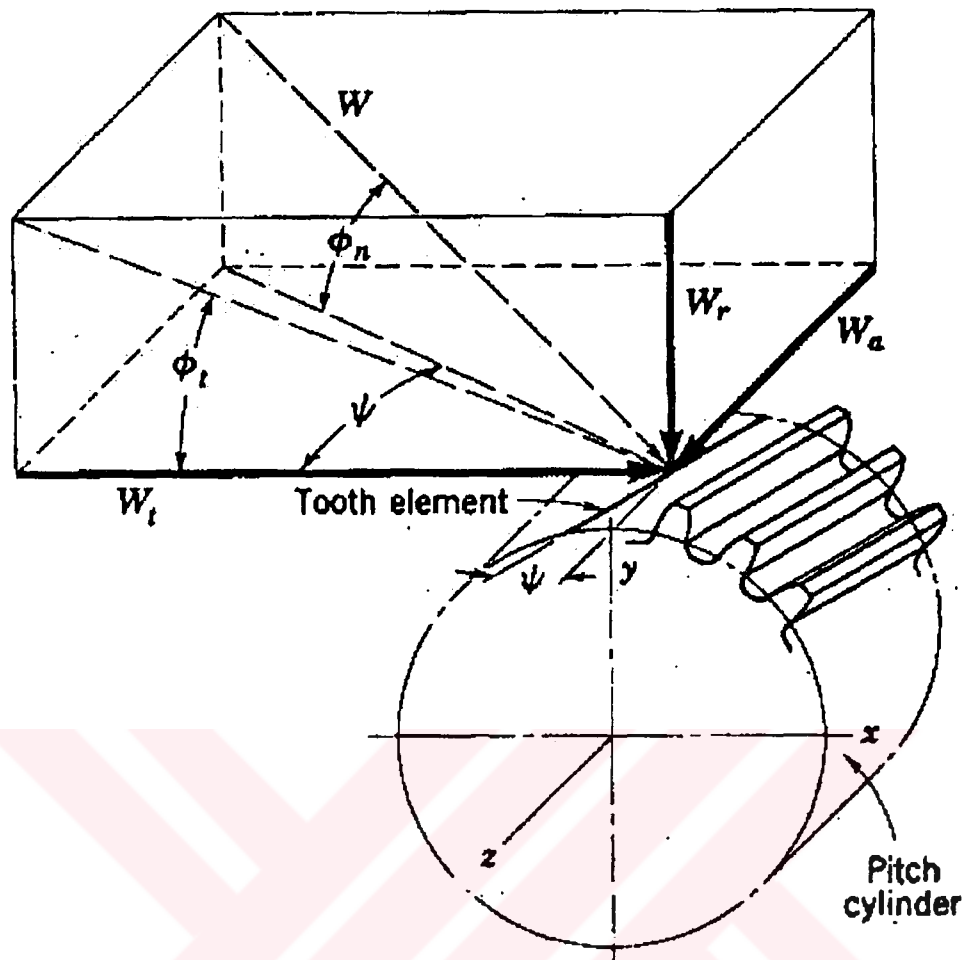


Figure 3.8 Tooth Forces Acting on a Right Hand Helical Gear

3.3.3 Strength Analysis

Bending and surface stresses in spur gears, i.e., equations (3.12) and (3.17) are also useful for helical gear design.

For helical gears the velocity factor is usually taken as

$$K_v = C_v = \sqrt{\frac{78}{78 + \sqrt{200V}}} \quad (3.31)$$

Geometry factors (J) for helical gears must account for the fact that contact occurs along a diagonal line across the tooth face and that it is usually dealt with the transverse pitch instead of the normal pitch. When the line of contact intersects with the tip of the tooth; then, the worst loading condition occurs.

The geometry factors J (for bending) for $\Phi_n=20^\circ$ can be found in Figure B1.

For helical gears, geometry factors I are calculated from the equation

$$I = \frac{\cos\Phi_t \cdot \sin\Phi_t}{2 \cdot m_N} \frac{m_G}{m_G + 1} \quad (3.32)$$

for external gears (use - for internal gears). In equation (3.32), Φ_t is the transverse pressure angle and m_N is the load sharing ratio and found from the equation

$$m_N = \frac{P_N}{0.95Z} \quad (3.33)$$

Where; P_N is the normal base pitch; it is related to the normal circular pitch P_n by the expression

$$P_N = P_n \cdot \cos \Phi_n \quad (3.34)$$

The quantity Z in equation (3.33) is the length of the line of action in the transverse plane and it can be found from

$$Z = \sqrt{(r_p + a)^2 - r_{bp}^2} + \sqrt{(r_G + a)^2 - r_{bG}^2} - (r_p + r_G) \sin \Phi_t \quad (3.35)$$

Where r_p and r_G are the pitch radii and r_{bp} and r_{bG} the base circle radii, respectively, for the pinion and gear. They can be found by the relations

$$\begin{aligned}
r_{bp} &= r_p \cdot \cos\Phi_t \\
r_{bg} &= r_g \cdot \cos\Phi_t \\
a &= m_n
\end{aligned}
\tag{3.36}$$

The tooth profiles are not conjugate below the base circle, and consequently, if either $\sqrt{(r_p + a)^2 - r_{bp}^2}$ or $\sqrt{(r_g + a)^2 - r_{bg}^2}$ is larger than $(r_p + r_g)\sin\Phi_t$, that term should be replaced by $(r_p + r_g)\sin\Phi_t$; i.e.

$$\begin{aligned}
\text{If } \sqrt{(r_p + a)^2 - r_{bp}^2} > (r_p + r_g)\sin\Phi_t, \quad Z &= \sqrt{(r_g + a)^2 - r_{bg}^2} \\
\text{If } \sqrt{(r_g + a)^2 - r_{bg}^2} > (r_p + r_g)\sin\Phi_t, \quad Z &= \sqrt{(r_p + a)^2 - r_{bp}^2}
\end{aligned}
\tag{3.37}$$

The modification and correction factors for helical gears are the same as for spur gears, except load distribution factors (K_m and C_m). Those factors can be found from table 3.5.

Table 3.5 Load Distribution Factor (K_m , C_m) for Helical Gears (40)

Characteristics of support	Face width, mm			
	0 to 50	150	225	400 up
Accurate mountings, small bearing clearances minimum deflection, precision G.	1.2	1.3	1.4	1.7
Less rigid mountings, less accurate gears, contact across full face	1.5	1.6	1.7	2.0
Accuracy and mounting such that less than full-face contact exists	Over		2.0	

Also the hardness ratio factor (C_H) can be found from Figure B1 in appendix. Where the factor K is the ratio of the Brinell hardness of the pinion to the Brinell hardness of the gear (use $C_H=1.0$ when $K < 1.2$)

3.4 GEAR MATERIALS

Most of the materials used for gears, other than the steels, are not subject to hardness control by heat-treatment. Materials that are commonly used for gears are steel, cast iron, bronze, or phenolic resins. Recently nylon, teflon, titanium, and sintered iron have also been used successfully. Available great variety of materials provides the designer with the opportunity of obtaining the optimum material for the requirement whether it has high strength, a long wear life, quietness of operation, or high reliability. Metallurgical quality is just as important as geometric quality.

Steel is the satisfactory material for many applications. Because, it has both high strength and low cost, gears are made of both plain carbon and alloy steels. The ultimate tensile strength of steel is a very direct function of the Brinell hardness.

Cast iron has good wear resistance. It is easy to cast and machine and transmits less noise than steel. That's why it is a very useful gear material.

When corrosion is a problem for gears, bronze may be used as material in production. They are quite useful for reducing friction and wear when the sliding velocity is high.

The raw steel used to make forgings for carburizing possible will not make a good gear if the machined forging has serious internal flaws before the final heat treating operation[41].

Non-metallic gears are mated with steel or cast iron gears to obtain the greatest load carrying capacity. To obtain good wear resistance, the metal gear should have a hardness of at least 300 BHN. Even though the strength of non metallic gear is much lower, it will carry almost as much load as a good cast iron or mild steel gear, because of the low modulus of elasticity. This low modulus permits the non-metallic gear to absorb the effects of tooth errors so that a dynamic load is not created.

CHAPTER 4

CAD OF SPUR AND HELICAL GEARS

4.1 INTRODUCTION

In this chapter a procedure for computer aided design of gear reduction unit will be presented. Also, criteria about the design and optimisation that used in the program will be examined and some case studies were presented in the next chapter.

4.2 COMPUTER AIDED DESIGN PROCEDURE

An automated design procedure is developed. This procedure will be used to provide a systematic approach to the design of gears.

Step1: Input power, speed ratio, factor of safety, tensile strength and Brinell hardness of material

Step2: Module, number of teeth of pinion and gear are taken automatically in the program as trial and error tools for constructing the design calculations.

Step3: Pitch diameter of pinion is calculated by using equation (3.1). Also, by using gear ratio as a multiplication factor, diameter of gear can be obtained.

Step4: Pitch line velocity of pinion is calculated from equation (3.6).

Step5: The transmitted load is calculated from equation (3.7).

Step 6: Velocity factor is determined through equations (3.8) to (3.11). For computer program, the modified form of velocity factor equation that is specified in equation (3.11) is used for obtaining the velocity factor.

Step 7: The AGMA geometry factor is selected directly from the related tables that were presented in the appendix by writing a subprogram.

Step 8: The endurance limit of rotating beam is determined from equation (3.14) and corrected by multiplying with some factors in equation (3.13).

Step 9: The gear factor of safety is calculated from equation (3.16).

Step10: The bending stress is calculated from equation (4.1)

$$\sigma = S_e / n_G \quad (4.1)$$

Step 11: From Lewis formula, bending stress is obtained by equation (3.12).

Rearranging equation (3.12) gives the face width as equation (4.2)

$$F = W_t / (\sigma \cdot K_v \cdot J \cdot m) \quad (4.2)$$

Step12: If calculated face width lies between $3\pi.m$ and $5\pi.m$ design is completed for fatigue strength. Otherwise some modifications are needed as

- a) To increase the number of teeth of pinion
- b) To increase the module
- c) To increase the strength of material

When designing requirement is satisfied, fatigue design for bending strength is completed. If it is necessary, it may continue to the next step.

Step 13: The surface fatigue strength is calculated from equation (3.20)

Step 14: Surface fatigue strength is modified by equation (3.21)

Step 15: Speed ratio is determined from subroutine DISORAN

Step 16: Geometry factor I is calculated from equation (3.19)

Step 17: The elastic coefficient is calculated from equation (3.18)

Step 18: The permissible transmitted load is determined from equation (3.23)

Step 19: The gear factor of safety n_G is the ratio of permissible transmitted load to the actual transmitted load

$$n_G = W_{tp} / W_t \quad (4.3)$$

Step 20: Factor of safety is calculated from step 19 must be equal (or greater than) to that given by equation (3.16)

Step 21: If n_G calculated from equation 4.3 is greater or equal to the given factor of safety (Eq. 3.16), design is completed for surface fatigue strength. Otherwise, some modifications are again needed such as

- a) To increase the number of teeth of the pinion
- b) To increase the module
- c) To increase the hardness

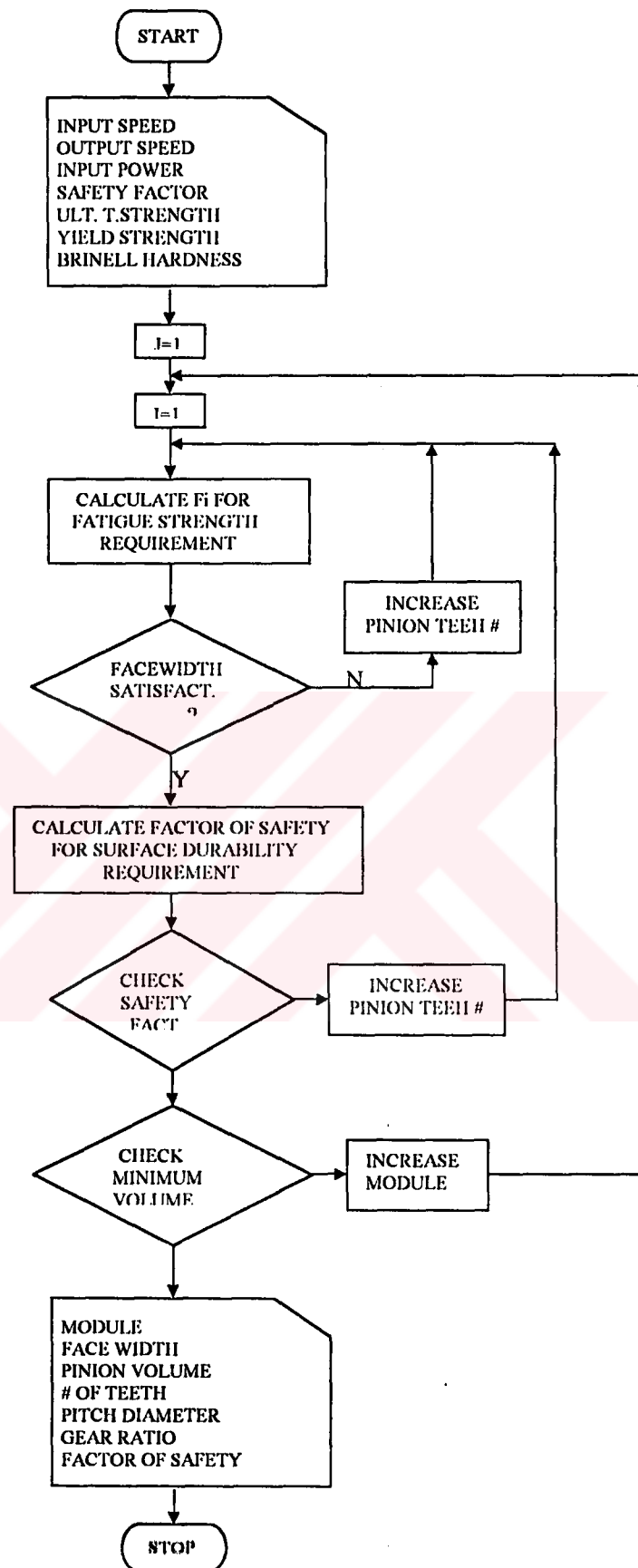


Figure 4.1: Simplified Flowchart of Computer Program for Sizing Gears

4.3 DESIGN PROCEDURE

Here, the aim is to determine parameters of gears in gear reduction units with minimum values of face widths, pitch diameters, modules and teeth numbers. In classical design of spur or helical gear transmission systems, the dimensions of the gears are assumed and then the strength calculations are performed.

Design of a single gear and the algorithm developed are thoroughly explained in the previous chapter. Here, the purpose is to discuss how to apply that algorithm to the design of gear reduction unit with prepared computer program.

The PC program has two options:

- Design of gear reduction unit.
- Optimisation of gear parameters for minimum volume.

A pair of gear design for reduction unit is performed by using AGMA standard. The modification factors are indicated in the program by writing a number of subprograms and they are used interactively by the program during performing design and optimisation processes. Those subprograms and their functions will be specified later.

At the first step of the program, it is necessary to input the transmitted power (HP), reliability, operational speed (input, output), helix angle (only for helical gears), pressure angle, gear material, both yield and ultimate strength of the material, life required, factor of safety, and mounting conditions.

After indicating those parameters, the amount of the transmitted load by each gear must be evaluated. Then the program starts to calculate the stress occurring at the gears by using necessary formulas for each value of design and optimisation loops present in the program. Those loops are:

- 1- Module loop
- 2- Pinion teeth loop

For module loop, standard practical values of the module were used.

For pinion teeth loop, value 18 is taken as minimum value and 300 as maximum value. Pinion teeth loop is constructed as interior loop in the programs.

Then, these stresses are taken into consideration to find the factor of safety in order to satisfy fatigue failure due to bending stress and surface fatigue failure due to surface durability.

When input parameters satisfy the design requirements for bending strength , those parameters are taken as input parameters for surface fatigue design. As a result of surface fatigue design factor of safety is indicated.

If calculated factor of safety for surface fatigue design is greater than or equal to the that of the fatigue strength design; calculated parameters are saved as outputs in OUTPUT.DAT file. This operation is continued up to the end of two loops and acceptable values are written in to the file output data file.

As a second procedure, it is necessary to optimise the gear parameters by maintaining the safety conditions.

The solution method for the optimisation in most cases, however, may employ the technique of the iteration in a systematic way until the parameter values which satisfy the objectives are found. Each parameter is varied through its probable range with an increment value which can give the required accuracy in results while other parameters are kept constant. In the program, all the possible solutions are checked and the results for all these runs are compared until finding the optimum condition. This requires long time unless the range of the parameter values are kept to minimum.

By these processes, all suitable conditions have been checked and minimum volume is indicated after each process completed. This minimum volume is being maintained until the program is terminated (loops completed).

Among different solutions presented in output data file, a solution which provides minimum volume for the specified gear reduction unit is selected.

4.4. DOUBLE STAGE GEAR REDUCTION UNIT

Mainly, double stage gear reduction unit can be considered as the combination of two different single stage gear reduction units. Their design principles are the same as the single stage gear reduction unit. These types of reduction units are used where large gear ratio is required..

In the design of double stage gear reduction units; center distance of the two stages are equalised for simplicity in production. In this study, after designing the gear reduction unit, smaller center distance is determined and this smaller center distance is increased by increasing the gear ratio of that stage in each run for obtaining the equal center distances.

4.4.1. Design Procedure for Double Stage Gear Reduction Unit

Step 1: Input the necessary information for the design similar to that of the single stage gear reduction unit.

Step 2: Calculate the overall gear ratio as the ratio of the output speed to the input speed.

Step 3: Calculate the gear ratio for the first and second stages respectively.

Step 4: Apply the steps that are necessary for designing the single stage gear reduction unit.

Step 5: Find out the gear parameters (output) for the first stage.

Step 6: Calculate the input parameters for the second stage.

Step 7: Apply the steps that are necessary for designing the single stage gear reduction unit.

Step 8: Find out the gear parameters (output) for the second stage.

Step 9: Compare the center distance of the two stages.

Step 10: If the center distance of the first stage is equal to that of the second stage in specified tolerance range; print the results. Otherwise, increase the gear ratio of the stage that has smaller center distance.

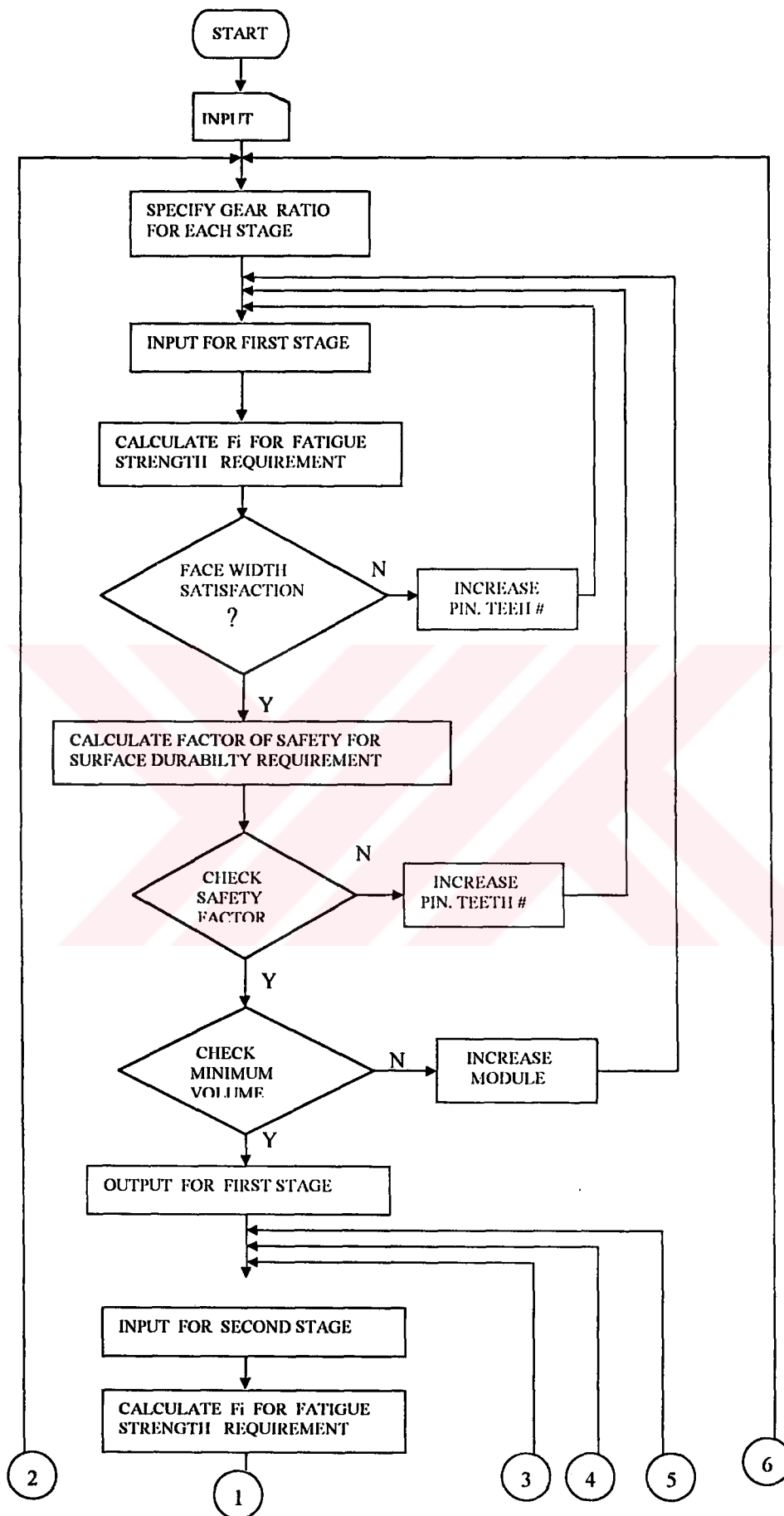
4.5 SUBROUTINES AND THEIR FUNCTIONS

In order to calculate (indicate) some factors (multipliers) there were some subroutines written for the main program. Those subprograms are called automatically when they are desired. Their application places are also shown in the flowcharts. These subroutines prevent complexity of the program.

The number names and functions of the subprograms were shown as below (Table 4.1).

Table 4.1 Subroutines and Their Functions

<u>No</u>	<u>Name</u>	<u>Functions</u>
1	AKA	Finds values of K_a
2	Function AMJ	Finds values of J
3	DISORAN	Calculates the gear ratio
4	AYK	Indicates value of K_f
5	LOAD DIST	Find the load distribution factor (cm)
6	KAYA	Find the (J) factor for helix gear design



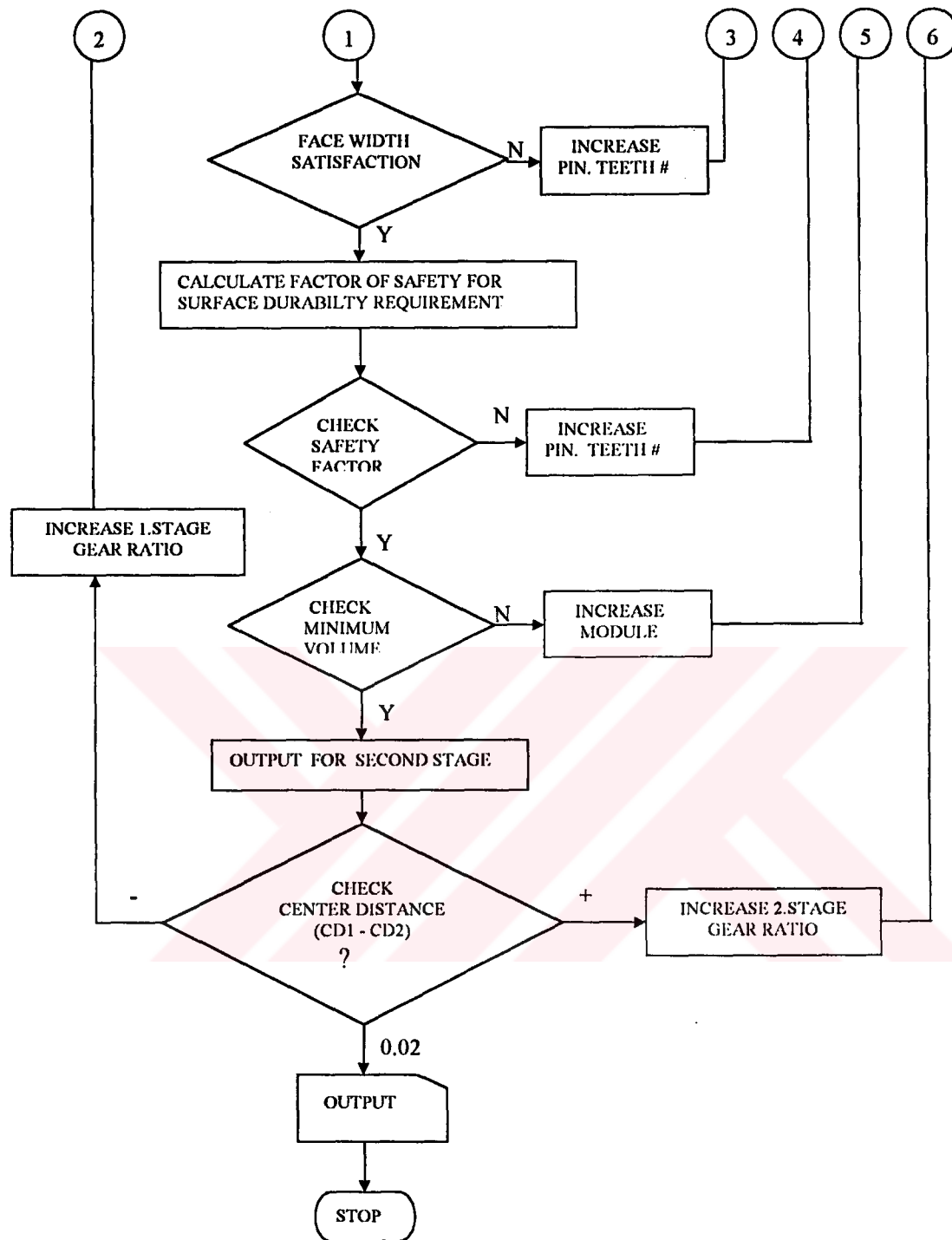


Figure 4.2: Flowchart for Designing the Double Stage Gear Reduction Unit

4.6 ESTIMATING GEAR SIZE FOR SPUR GEAR

While designing a gear set to determine the reliability corresponding to a specified life, or to calculate the factor of safety guarding against the failures, it is necessary to know the dimensions of the gears and the materials used for production. In this section, a preliminary estimate of the size of gears required to carry a given load is presented.

The choice of a face width in the range $3p \leq F \leq 5p$ was used as the design approach of gear sets. Gear sets having face widths greater than the five times the circular pitch are quite likely to have a non-uniform distribution of the load across the face of the tooth because of the torsional deflection of the gear and shaft, because of the machining inaccuracies, and because of the necessity of maintaining very accurate and rigid bearing mountings.

If the face width is less than three times the circular pitch, a larger gear is needed for carrying the larger load per unit of face width. Large gears require more space in the gear reduction unit and makes the machine bigger. Large gears require larger machines to generate teeth, and this increases the cost. For these reasons, a face width of three times the circular pitch is recommended for designing the gear pairs.

Both the transmitted load and the velocity depend, directly or indirectly, on the module m . That's why, the gear size is obtained using iteration techniques.

4.7 TOOTH PROPORTIONS FOR HELICAL GEARS

There is no standard for the proportions of the helical gear teeth except for fine-pitch gears. One reason for this is that it is cheaper to change the design slightly than it is to purchase special tooling[40].

Tooth proportions should be based on a normal pressure angle of 20° in general. The dimensions of tooth were calculated (should be calculated) by using the normal module. These proportions are suitable for helix angle from 0° to 30° , and all helix angles may be cut with the same hob. For the hob and the gear the normal module must be the same.

An optimal set of proportions can be based on a transverse pressure angle of 20° . For these reason the helix angles are generally restricted to 15° , 23° , 30° or 45° . Greater angles are not recommended.

Many authorities recommend that the face width of helical gears should be at least two times the axial pitch ($F=2P_x$) to obtain helical-gear action.

CHAPTER 5

CASE STUDIES

5.1 INTRODUCTION

In his chapter, few examples that are related to the single stage helical gear reduction units of MURAT-131, WOLKSWAGON automobiles were presented for testing the output of the program for minimum volume design criteria. For spur gears, reduction unit of BLACK and DECKER hammer drill was tested. Also, for double stage gear reduction unit, results are compared with gear reduction units produced by YILMAZ REDÜKTÖR. All dimensions presented in tables are in mm. unless stated.

5.2 GEAR REDUCTION UNIT OF MURAT-131 AUTOMOBILE

The data of the MURAT-131 were taken from the catalogue. Cross- section views of MURAT-131 gearbox is shown in figure 5.1. The related data about the gears used in the gear box are tabulated in Table.5.1. The program was executed for minimum volume case by taking the original gear-box and material parameters.

For the minimum volume case , inputs and outputs data are shown in Table 5.2. Some acceptable output data are also presented in Table 5.4.

Table 5.1.The Dimensions of the Gears Used in a MURAT-131 Automobile Gear-box

Name	First Gear	Second Gear	Third Gear	Fourth Gear
Number of Teeth	28	33	27	21
Normal Module	2.5	2.5	2.5	2.5
Transverse Module	2.9166	2.8584	2.8584	2.9166
Pitch Diameter	81.66	94.327	77.1764	61.248
Outside Diameter	87.10	99.10	82.60	69.30
Root Diameter	73.70	86.476	69.20	55.90
Base Diameter	78.18	87.087	74.0088	58.6370
Normal Press. Ang.	14.5	20	14.5	14.5
Trans. Press. Ang.	16°47'	22°35'	16°28'	16°47'
Pitch Helix Ang.	31°	29°	29°	31°
Base Helix Ang	29°54'	27°60'	27°59'	29°54'
Lead	426.98	534.60	427.40	320.24
Addendum	2.75	2.42	2.75	4.09
No. Of Teeth of Meshing Gear	18	14	20	24
Center Distance	68.00	68.00	68.00	68.00
Theoretical Center D.	67.08	67.17	67.17	65.62
Velocity Ratio	1.555	2.357	1.350	0.875

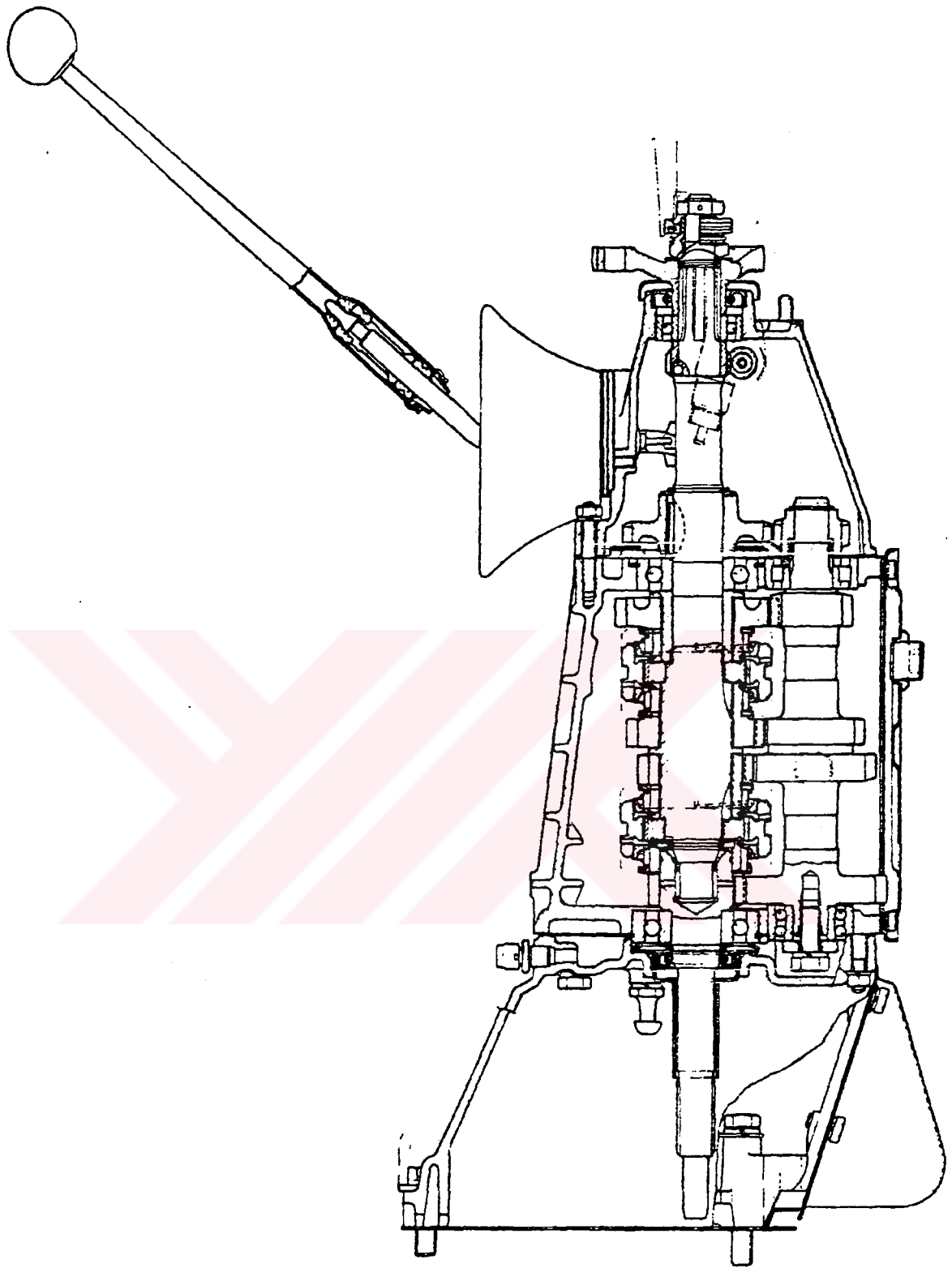


Figure 5.1 Gearbox of a Murat 131 Automobile

Table 5.2. Input and Calculated Data of First Gear Pair (Constant Mesh) Used in
MURAT-131 Gear-box for Single Stage Helical GRU

INPUT DATA

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	70.00
The input speed value as (RPM).....:	5250.00
The output speed value as (RPM).....:	3376.00
Helix angle as (Degree).....:	30.00
Cycles of life.....:	1000000

OUTPUT OF THE PROGRAM

The Smallest Volume (Cubic Cent.).....:	40.00
Module.....:	2.75
Face Width	29.93
Center Distance	52.70
Number of Teeth of Pinion.....:	15
Number of Teeth of Gear.....:	23
Pitch Diameter of Pinion	41.25
Pitch Diameter of Gear.....:	64.15
Gear Ratio.....:	1.55
Factor Of Safety.....:	3.25

Table 5.3. Input and Calculated Data of Second Gear Pair Used in MURAT-131
Gear-box for Single Stage Helical GRU

INPUT DATA

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	70.00
The input speed value as (RPM).....:	3376.00
The output speed value as (RPM).....:	1432.33
Helix angle as (Degree).....:	30.00
Cycles of life.....:	1000000

OUTPUT OF THE PROGRAM

The Smallest Volume (Cubic Cent.).....:	71.83
Module.....:	3.50
Face Width	38.09
Center Distance	82.25
Number of Teeth of Pinion.....:	14
Number of Teeth of Gear.....:	32
Pitch Diameter of Pinion	49.00
Pitch Diameter of Gear.....:	115.49
Gear Ratio.....:	2.35
Factor Of Safety.....:	4.05

**Table 5.4. Calculated Data of First Gear Pair Used in MURAT-131 Gear-box for
Single Stage Helical GRU**

VOLUME	CENT. D.	MODULE	NP	NG	DP	DG	FACE	S. FACT.
138.47	114.98	2.00	45	69	90.00	139.96	21.77	3.20
144.69	117.53	2.00	46	71	92.00	143.07	21.77	3.25
151.05	120.09	2.00	47	73	94.00	146.18	21.77	3.31
157.54	122.64	2.00	48	74	96.00	149.29	21.77	3.36
164.18	125.20	2.00	49	76	98.00	152.40	21.77	3.42
170.95	127.75	2.00	50	77	100.00	155.51	21.77	3.47
87.62	86.23	2.25	30	46	67.50	104.97	24.49	3.22
93.56	89.11	2.25	31	48	69.75	108.47	24.49	3.31
99.70	91.98	2.25	32	49	72.00	111.97	24.49	3.39
106.02	94.86	2.25	33	51	74.25	115.47	24.49	3.48
112.55	97.73	2.25	34	52	76.50	118.96	24.49	3.56
119.27	100.61	2.25	35	54	78.75	122.46	24.49	3.64
126.18	103.48	2.25	36	55	81.00	125.96	24.49	3.72
133.29	106.36	2.25	37	57	83.25	129.46	24.49	3.81
140.59	109.23	2.25	38	59	85.50	132.96	24.49	3.89
243.40	143.72	2.25	50	77	112.50	174.95	24.49	4.82
253.23	146.60	2.25	51	79	114.75	178.45	24.49	4.89
263.26	149.47	2.25	52	80	117.00	181.95	24.49	4.97
58.90	67.07	2.50	21	32	52.50	81.64	27.21	3.26
64.64	70.27	2.50	22	34	55.00	85.53	27.21	3.38
70.65	73.46	2.50	23	35	57.50	89.42	27.21	3.50
76.93	76.65	2.50	24	37	60.00	93.31	27.21	3.63
83.47	79.85	2.50	25	38	62.50	97.19	27.21	3.75
90.28	83.04	2.50	26	40	65.00	101.08	27.21	3.86
97.36	86.23	2.50	27	41	67.50	104.97	27.21	3.98
104.71	89.43	2.50	28	43	70.00	108.86	27.21	4.10
112.32	92.62	2.50	29	45	72.50	112.74	27.21	4.21
40.00	52.70	2.75	15	23	41.25	64.15	29.93	3.25
45.51	56.21	2.75	16	24	44.00	68.42	29.93	3.42
51.37	59.73	2.75	17	26	46.75	72.70	29.93	3.59
57.59	63.24	2.75	18	27	49.50	76.98	29.93	3.76
64.17	66.75	2.75	19	29	52.25	81.25	29.93	3.93
71.10	70.27	2.75	20	31	55.00	85.53	29.93	4.09
78.39	73.78	2.75	21	32	57.75	89.81	29.93	4.26
86.03	77.29	2.75	22	34	60.50	94.08	29.93	4.42
94.03	80.80	2.75	23	35	63.25	98.36	29.93	4.57
102.39	84.32	2.75	24	37	66.00	102.64	29.93	4.73
45.23	53.66	3.00	14	21	42.00	65.31	32.65	3.92
51.93	57.49	3.00	15	23	45.00	69.98	32.65	4.15
59.08	61.32	3.00	16	24	48.00	74.64	32.65	4.37
66.69	65.15	3.00	17	26	51.00	79.31	32.65	4.58
74.77	68.99	3.00	18	27	54.00	83.98	32.65	4.80
83.31	72.82	3.00	19	29	57.00	88.64	32.65	5.01
282.70	134.14	3.00	35	54	105.00	163.28	32.65	8.12
299.09	137.98	3.00	36	55	108.00	167.95	32.65	8.30
315.94	141.81	3.00	37	57	111.00	172.62	32.65	8.47
333.24	145.64	3.00	38	59	114.00	177.28	32.65	8.65
351.01	149.47	3.00	39	60	117.00	181.95	32.65	8.83
71.83	62.60	3.50	14	21	49.00	76.20	38.09	6.04
82.46	67.07	3.50	15	23	52.50	81.64	38.09	6.39

**Table 5.5. Calculated Data of Second Gear Pair Used in MURAT-131 Gear-box for
Single Stage Helical GRU**

VOLUME	CENT.D.	MODULE	NP	NG	DP	DG	FACE	S.FACT.
154.39	142.67	2.50	34	80	85.00	200.34	27.21	3.22
163.60	146.87	2.50	35	82	87.50	206.24	27.21	3.30
102.39	110.78	2.75	24	56	66.00	155.56	29.93	3.18
111.10	115.40	2.75	25	58	68.75	162.04	29.93	3.29
120.16	120.01	2.75	26	61	71.50	168.53	29.93	3.40
129.59	124.63	2.75	27	63	74.25	175.01	29.93	3.50
139.36	129.24	2.75	28	65	77.00	181.49	29.93	3.60
149.49	133.86	2.75	29	68	79.75	187.97	29.93	3.71
159.98	138.48	2.75	30	70	82.50	194.45	29.93	3.81
170.83	143.09	2.75	31	73	85.25	200.93	29.93	3.91
182.02	147.71	2.75	32	75	88.00	207.42	29.93	4.01
74.77	90.64	3.00	18	42	54.00	127.28	32.65	3.22
83.31	95.67	3.00	19	44	57.00	134.35	32.65	3.36
92.31	100.71	3.00	20	47	60.00	141.42	32.65	3.51
101.77	105.75	3.00	21	49	63.00	148.49	32.65	3.65
111.70	110.78	3.00	22	51	66.00	155.56	32.65	3.79
122.08	115.82	3.00	23	54	69.00	162.63	32.65	3.93
132.93	120.85	3.00	24	56	72.00	169.70	32.65	4.06
144.24	125.89	3.00	25	58	75.00	176.77	32.65	4.20
156.01	130.92	3.00	26	61	78.00	183.85	32.65	4.33
168.24	135.96	3.00	27	63	81.00	190.92	32.65	4.47
180.93	140.99	3.00	28	65	84.00	197.99	32.65	4.60
194.08	146.03	3.00	29	68	87.00	205.06	32.65	4.73
71.83	82.25	3.50	14	32	49.00	115.49	38.09	4.05
82.46	88.12	3.50	15	35	52.50	123.74	38.09	4.28
93.82	94.00	3.50	16	37	56.00	131.99	38.09	4.51
105.91	99.87	3.50	17	40	59.50	140.24	38.09	4.74
118.74	105.75	3.50	18	42	63.00	148.49	38.09	4.97
132.29	111.62	3.50	19	44	66.50	156.74	38.09	5.19
146.59	117.49	3.50	20	47	70.00	164.99	38.09	5.41
161.61	123.37	3.50	21	49	73.50	173.24	38.09	5.62
177.37	129.24	3.50	22	51	77.00	181.49	38.09	5.84
193.86	135.12	3.50	23	54	80.50	189.74	38.09	6.05
211.09	140.99	3.50	24	56	84.00	197.99	38.09	6.26
229.04	146.87	3.50	25	58	87.50	206.24	38.09	6.47
107.22	94.00	4.00	14	32	56.00	131.99	43.53	5.90
123.08	100.71	4.00	15	35	60.00	141.42	43.53	6.24
140.04	107.42	4.00	16	37	64.00	150.85	43.53	6.57
158.09	114.14	4.00	17	40	68.00	160.28	43.53	6.90
177.24	120.85	4.00	18	42	72.00	169.70	43.53	7.22
197.48	127.57	4.00	19	44	76.00	179.13	43.53	7.55
218.81	134.28	4.00	20	47	80.00	188.56	43.53	7.86
241.24	140.99	4.00	21	49	84.00	197.99	43.53	8.18
264.76	147.71	4.00	22	51	88.00	207.42	43.53	8.49

Table 5.6. Input and Calculated Data of Third Gear Pair Used in MURAT-131 Gear-box for Single Stage Helical GRU

INPUT DATA	

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	70.00
The input speed value as (RPM).....:	3376.00
The output speed value as (RPM).....:	2500.00
Helix angle as (Degree).....:	30.00
Cycles of life.....:	1000000
OUTPUT OF THE PROGRAM	

The Smallest Volume (Cubic Cent.).....:	71.83
Module.....:	3.50
Face Width	38.09
Center Distance	57.58
Number of Teeth of Pinion.....:	14
Number of Teeth of Gear.....:	18
Pitch Diameter of Pinion	49.00
Pitch Diameter of Gear.....:	66.17
Gear Ratio.....:	1.35
Factor Of Safety.....:	4.05

Table 5.7. Input and Calculated Data of Fourth Gear Pair Used in MURAT-131 Gear-box for Single Stage Helical GRU

INPUT DATA	

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	70.00
The input speed value as (RPM).....:	3376.00
The output speed value as (RPM).....:	3858.00
Helix angle as (Degree).....:	30.00
Cycles of life.....:	1000000
OUTPUT OF THE PROGRAM	

The Smallest Volume (Cubic Cent.).....:	74.77
Module.....:	3.00
Face Width	32.65
Center Distance	57.86
Number of Teeth of Pinion.....:	18
Number of Teeth of Gear.....:	20
Pitch Diameter of Pinion	54.00
Pitch Diameter of Gear.....:	61.71
Gear Ratio.....:	1.14
Factor Of Safety.....:	3.22

**Table 5.8. Calculated Data of Third Gear Pair Used in MURAT-131 Gear-box for
Single Stage Helical GRU**

VOLUME	CENT.D.	MODULE	NP	NG	DP	DG	FACE	S.FACT.
273.48	140.14	2.25	53	71	119.25	161.04	24.49	3.42
283.90	142.79	2.25	54	72	121.50	164.07	24.49	3.47
294.51	145.43	2.25	55	74	123.75	167.11	24.49	3.52
305.32	148.08	2.25	56	75	126.00	170.15	24.49	3.57
154.39	99.89	2.50	34	45	85.00	114.78	27.21	3.22
163.60	102.83	2.50	35	47	87.50	118.16	27.21	3.30
173.08	105.77	2.50	36	48	90.00	121.54	27.21	3.37
182.83	108.71	2.50	37	49	92.50	124.91	27.21	3.45
192.85	111.64	2.50	38	51	95.00	128.29	27.21	3.52
203.13	114.58	2.50	39	52	97.50	131.66	27.21	3.60
213.68	117.52	2.50	40	54	100.00	135.04	27.21	3.67
102.39	77.56	2.75	24	32	66.00	89.13	29.93	3.18
111.10	80.79	2.75	25	33	68.75	92.84	29.93	3.29
120.16	84.03	2.75	26	35	71.50	96.55	29.93	3.40
129.59	87.26	2.75	27	36	74.25	100.27	29.93	3.50
139.36	90.49	2.75	28	37	77.00	103.98	29.93	3.60
149.49	93.72	2.75	29	39	79.75	107.69	29.93	3.71
159.98	96.95	2.75	30	40	82.50	111.41	29.93	3.81
74.77	63.46	3.00	18	24	54.00	72.92	32.65	3.22
83.31	66.99	3.00	19	25	57.00	76.97	32.65	3.36
92.31	70.51	3.00	20	27	60.00	81.02	32.65	3.51
101.77	74.04	3.00	21	28	63.00	85.08	32.65	3.65
111.70	77.56	3.00	22	29	66.00	89.13	32.65	3.79
122.08	81.09	3.00	23	31	69.00	93.18	32.65	3.93
132.93	84.61	3.00	24	32	72.00	97.23	32.65	4.06
144.24	88.14	3.00	25	33	75.00	101.28	32.65	4.20
156.01	91.67	3.00	26	35	78.00	105.33	32.65	4.33
168.24	95.19	3.00	27	36	81.00	109.38	32.65	4.47
180.93	98.72	3.00	28	37	84.00	113.43	32.65	4.60
194.08	102.24	3.00	29	39	87.00	117.48	32.65	4.73
71.83	57.58	3.50	14	18	49.00	66.17	38.09	4.05
82.46	61.70	3.50	15	20	52.50	70.90	38.09	4.28
93.82	65.81	3.50	16	21	56.00	75.62	38.09	4.51
105.91	69.92	3.50	17	22	59.50	80.35	38.09	4.74
118.74	74.04	3.50	18	24	63.00	85.08	38.09	4.97
132.29	78.15	3.50	19	25	66.50	89.80	38.09	5.19
146.59	82.26	3.50	20	27	70.00	94.53	38.09	5.41
161.61	86.38	3.50	21	28	73.50	99.25	38.09	5.62
177.37	90.49	3.50	22	29	77.00	103.98	38.09	5.84
193.86	94.60	3.50	23	31	80.50	108.71	38.09	6.05
211.09	98.72	3.50	24	32	84.00	113.43	38.09	6.26
229.04	102.83	3.50	25	33	87.50	118.16	38.09	6.47
247.73	106.94	3.50	26	35	91.00	122.89	38.09	6.67
267.15	111.06	3.50	27	36	94.50	127.61	38.09	6.88
287.31	115.17	3.50	28	37	98.00	132.34	38.09	7.08
308.20	119.28	3.50	29	39	101.50	137.07	38.09	7.28
329.82	123.40	3.50	30	40	105.00	141.79	38.09	7.48
352.18	127.51	3.50	31	41	108.50	146.52	38.09	7.68
375.26	131.62	3.50	32	43	112.00	151.24	38.09	7.87
399.08	135.74	3.50	33	44	115.50	155.97	38.09	8.06
423.64	139.85	3.50	34	45	119.00	160.70	38.09	8.26

**Table 5.9. Calculated Data of Fourth Gear Pair Used in MURAT-131 Gear-box for
Single Stage Helical GRU**

VOLUME	CENT.D.	MODULE	NP	NG	DP	DG	FACE	S.FACT.
154.39	91.07	2.50	34	38	85.00	97.14	27.21	3.22
163.60	93.75	2.50	35	40	87.50	100.00	27.21	3.30
173.08	96.43	2.50	36	41	90.00	102.86	27.21	3.37
182.83	99.11	2.50	37	42	92.50	105.71	27.21	3.45
192.85	101.79	2.50	38	43	95.00	108.57	27.21	3.52
203.13	104.46	2.50	39	44	97.50	111.43	27.21	3.60
213.68	107.14	2.50	40	45	100.00	114.29	27.21	3.67
224.50	109.82	2.50	41	46	102.50	117.14	27.21	3.74
235.59	112.50	2.50	42	48	105.00	120.00	27.21	3.82
246.94	115.18	2.50	43	49	107.50	122.86	27.21	3.89
258.56	117.86	2.50	44	50	110.00	125.71	27.21	3.96
270.44	120.54	2.50	45	51	112.50	128.57	27.21	4.03
282.60	123.21	2.50	46	52	115.00	131.43	27.21	4.10
295.02	125.89	2.50	47	53	117.50	134.29	27.21	4.17
307.70	128.57	2.50	48	54	120.00	137.14	27.21	4.24
102.39	70.71	2.75	24	27	66.00	75.43	29.93	3.18
111.10	73.66	2.75	25	28	68.75	78.57	29.93	3.29
120.16	76.61	2.75	26	29	71.50	81.71	29.93	3.40
129.59	79.55	2.75	27	30	74.25	84.86	29.93	3.50
139.36	82.50	2.75	28	32	77.00	88.00	29.93	3.60
149.49	85.45	2.75	29	33	79.75	91.14	29.93	3.71
159.98	88.39	2.75	30	34	82.50	94.29	29.93	3.81
170.83	91.34	2.75	31	35	85.25	97.43	29.93	3.91
182.02	94.29	2.75	32	36	88.00	100.57	29.93	4.01
193.58	97.23	2.75	33	37	90.75	103.71	29.93	4.11
74.77	57.86	3.00	18	20	54.00	61.71	32.65	3.22
83.31	61.07	3.00	19	21	57.00	65.14	32.65	3.36
92.31	64.29	3.00	20	22	60.00	68.57	32.65	3.51
101.77	67.50	3.00	21	24	63.00	72.00	32.65	3.65
111.70	70.71	3.00	22	25	66.00	75.43	32.65	3.79
122.08	73.93	3.00	23	26	69.00	78.86	32.65	3.93
132.93	77.14	3.00	24	27	72.00	82.29	32.65	4.06
144.24	80.36	3.00	25	28	75.00	85.71	32.65	4.20
156.01	83.57	3.00	26	29	78.00	89.14	32.65	4.33
168.24	86.79	3.00	27	30	81.00	92.57	32.65	4.47
180.93	90.00	3.00	28	32	84.00	96.00	32.65	4.60
194.08	93.21	3.00	29	33	87.00	99.43	32.65	4.73
118.74	67.50	3.50	18	20	63.00	72.00	38.09	4.97
132.29	71.25	3.50	19	21	66.50	76.00	38.09	5.19
146.59	75.00	3.50	20	22	70.00	80.00	38.09	5.41
161.61	78.75	3.50	21	24	73.50	84.00	38.09	5.62
177.37	82.50	3.50	22	25	77.00	88.00	38.09	5.84
193.86	86.25	3.50	23	26	80.50	92.00	38.09	6.05
211.09	90.00	3.50	24	27	84.00	96.00	38.09	6.26
229.04	93.75	3.50	25	28	87.50	100.00	38.09	6.47
247.73	97.50	3.50	26	29	91.00	104.00	38.09	6.67
267.15	101.25	3.50	27	30	94.50	108.00	38.09	6.88
287.31	105.00	3.50	28	32	98.00	112.00	38.09	7.08
308.20	108.75	3.50	29	33	101.50	116.00	38.09	7.28
329.82	112.50	3.50	30	34	105.00	120.00	38.09	7.48
352.18	116.25	3.50	31	35	108.50	124.00	38.09	7.68
375.26	120.00	3.50	32	36	112.00	128.00	38.09	7.87

5.3 GEAR REDUCTION UNIT OF BLACK & DECKER HAND DRILL

Gears used in Black and Decker Hand Drill are spur gears. The original parameters of gears have been taken from a dismantling two speed gear-box which is present in the laboratory. Those data are shown in Table 5.10.

For the minimum volume case, input and output data of first gear, and second gear are shown in related tables.

Table 5.10. The Dimensions of the Gears Used in Black & Decker Hammer Drill Gear-Box

Name	First Gear	Second Gear
No. Of Teeth	33	41
No. of Teeth of Mesh Gear	18	10
Module	0.75	0.75
Face width	8.0	8.8
Pitch Diameter	24.75	30.75
Outside Diameter	26.25	32.25
Center Distance	19.125	19.125
Press. Angle	20°	20°

**Table 5.11. Input and Calculated Data of First Gear Pair Used in Black & Decker
Hammer Drill Gear-box for Single Stage Helical GRU**

INPUT DATA	
TYPE OF THE MATERIAL.....:	Steel
ULTIMATE TENSILE STRENGTH(MPa).....:	600.00
THE BRINELL HARDNESS NUMBER	322.00
THE INPUT POWER(HORSE POWER).....:	0.60
THE INPUT SPEED VALUE (RPM).....:	5125.00
THE GEAR RATIO.....:	1.83
OUTPUT OF THE PROGRAM	
THE SMALLEST VOLUME (Cubic Centimeter)...	1.47
MODULE.....:	0.75
THE FACE WIDTH	9.23
WHOLE DEPTH OF CUT	1.62
THE NUMBER OF PINION TEETH	19
THE NUMBER OF GEAR TEETH	34
PINION PITCH DIAMETER	14.25
GEAR PITCH DIAMETER	26.08
PINION OUTSIDE DIAMETER	15.75
GEAR OUTSIDE DIAMETER	27.58
FACTOR OF SAFETY FOR BENDING.....:	3.41
FACTOR OF SAFETY FOR SURFACE DURAB.....:	3.48

**Table 5.12. Input and Calculated Data of Second Gear Pair Used in Black & Decker
Hammer Drill Gear-box for Single Stage Helical GRU**

INPUT DATA	
TYPE OF THE MATERIAL.....:	Steel
ULTIMATE TENSILE STRENGTH(MPa).....:	500.00
THE BRINELL HARDNESS NUMBER	322.00
THE INPUT POWER(HORSE POWER).....:	0.60
THE INPUT SPEED VALUE (RPM).....:	5125.00
THE GEAR RATIO.....:	4.10
OUTPUT OF THE PROGRAM	
THE SMALLEST VOLUME (Cubic Centimeter)...	1.25
MODULE.....:	1.00
THE FACE WIDTH	13.11
WHOLE DEPTH OF CUT	2.17
THE NUMBER OF PINION TEETH	11
THE NUMBER OF GEAR TEETH	45
PINION PITCH DIAMETER	11.00
GEAR PITCH DIAMETER	45.10
PINION OUTSIDE DIAMETER	13.00
GEAR OUTSIDE DIAMETER	47.10
FACTOR OF SAFETY FOR BENDING.....:	3.41
FACTOR OF SAFETY FOR SURFACE DURAB.....:	3.73

Table 5.13. Calculated Data of First Gear Pair Used in Black & Decker Hammer Drill

Gear-box for Single Stage Spur GRU

VOLUME	CENT.D.	MODULE	NP	DP	DG	FACE	B.F.S	S.F.S
2.41	28.30	0.50	40.00	20.00	36.61	7.68	3.41	5.58
2.47	29.01	0.50	41.00	20.50	37.52	7.48	3.41	5.70
2.52	29.72	0.50	42.00	21.00	38.44	7.28	3.41	5.81
2.58	30.43	0.50	43.00	21.50	39.35	7.10	3.41	5.93
2.63	31.13	0.50	44.00	22.00	40.27	6.92	3.41	6.04
2.69	31.84	0.50	45.00	22.50	41.18	6.75	3.41	6.15
2.74	32.55	0.50	46.00	23.00	42.10	6.60	3.41	6.27
2.80	33.26	0.50	47.00	23.50	43.01	6.45	3.41	6.39
2.86	33.96	0.50	48.00	24.00	43.93	6.31	3.41	6.51
2.91	34.67	0.50	49.00	24.50	44.84	6.18	3.41	6.63
2.97	35.38	0.50	50.00	25.00	45.76	6.04	3.41	6.74
3.03	36.09	0.50	51.00	25.50	46.67	5.92	3.41	6.87
3.08	36.79	0.50	52.00	26.00	47.59	5.81	3.41	6.99
3.14	37.50	0.50	53.00	26.50	48.50	5.70	3.41	7.11
3.20	38.21	0.50	54.00	27.00	49.42	5.59	3.41	7.23
3.26	38.92	0.50	55.00	27.50	50.33	5.49	3.41	7.35
3.32	39.63	0.50	56.00	28.00	51.25	5.39	3.41	7.47
3.37	40.33	0.50	57.00	28.50	52.17	5.29	3.41	7.59
3.43	41.04	0.50	58.00	29.00	53.08	5.20	3.41	7.71
3.49	41.75	0.50	59.00	29.50	54.00	5.10	3.41	7.82
3.55	42.46	0.50	60.00	30.00	54.91	5.02	3.41	7.94
3.61	43.16	0.50	61.00	30.50	55.83	4.93	3.41	8.06
3.66	43.87	0.50	62.00	31.00	56.74	4.85	3.41	8.18
3.72	44.58	0.50	63.00	31.50	57.66	4.78	3.41	8.30
1.47	20.17	0.75	19.00	14.25	26.08	9.23	3.41	3.48
1.52	21.23	0.75	20.00	15.00	27.46	8.62	3.41	3.60
1.58	22.29	0.75	21.00	15.75	28.83	8.09	3.41	3.71
1.63	23.35	0.75	22.00	16.50	30.20	7.61	3.41	3.81
1.68	24.41	0.75	23.00	17.25	31.57	7.18	3.41	3.92

Table 5.14. Calculated Data of Second Gear Pair Used in Black & Decker Hammer

Drill Gear-box for Single Stage Spur GRU

VOLUME	CENT.D.	MODULE	NP	DP	DG	FACE	B.F.S.	S.F.S.
4.93	72.68	0.50	57.00	28.50	116.85	7.72	3.41	13.78
4.99	73.95	0.50	58.00	29.00	118.90	7.56	3.41	13.94
5.06	75.22	0.50	59.00	29.50	120.95	7.40	3.41	14.11
5.13	76.50	0.50	60.00	30.00	123.00	7.25	3.41	14.27
5.19	77.77	0.50	61.00	30.50	125.05	7.11	3.41	14.43
5.26	79.05	0.50	62.00	31.00	127.10	6.97	3.41	14.60
5.33	80.32	0.50	63.00	31.50	129.15	6.83	3.41	14.76
5.39	81.60	0.50	64.00	32.00	131.20	6.70	3.41	14.93
5.46	82.88	0.50	65.00	32.50	133.25	6.58	3.41	15.09
5.52	84.15	0.50	66.00	33.00	135.30	6.46	3.41	15.24
5.59	85.42	0.50	67.00	33.50	137.35	6.34	3.41	15.40
5.65	86.70	0.50	68.00	34.00	139.40	6.22	3.41	15.55
5.71	87.97	0.50	69.00	34.50	141.45	6.11	3.41	15.71
5.78	89.25	0.50	70.00	35.00	143.50	6.01	3.41	15.87
5.86	90.53	0.50	71.00	35.50	145.55	5.92	3.41	16.08
5.95	91.80	0.50	72.00	36.00	147.60	5.84	3.41	16.29
6.03	93.07	0.50	73.00	36.50	149.65	5.76	3.41	16.49
6.11	94.35	0.50	74.00	37.00	151.70	5.68	3.41	16.70
6.19	95.63	0.50	75.00	37.50	153.75	5.61	3.41	16.90
6.28	96.90	0.50	76.00	38.00	155.80	5.54	3.41	17.12
6.36	98.17	0.50	77.00	38.50	157.85	5.47	3.41	17.33
6.45	99.45	0.50	78.00	39.00	159.90	5.40	3.41	17.54
6.54	100.72	0.50	79.00	39.50	161.95	5.33	3.41	17.75
6.62	102.00	0.50	80.00	40.00	164.00	5.27	3.41	17.97
6.71	103.28	0.50	81.00	40.50	166.05	5.21	3.41	18.18
6.79	104.55	0.50	82.00	41.00	168.10	5.14	3.41	18.39
6.88	105.82	0.50	83.00	41.50	170.15	5.08	3.41	18.59
6.96	107.10	0.50	84.00	42.00	172.20	5.02	3.41	18.80
7.05	108.38	0.50	85.00	42.50	174.25	4.97	3.41	19.01
7.13	109.65	0.50	86.00	43.00	176.30	4.91	3.41	19.22
7.22	110.92	0.50	87.00	43.50	178.35	4.86	3.41	19.42
7.30	112.20	0.50	88.00	44.00	180.40	4.80	3.41	19.63
7.38	113.47	0.50	89.00	44.50	182.45	4.75	3.41	19.84
2.40	42.08	0.75	22.00	16.50	67.65	11.24	3.41	7.01
2.50	43.99	0.75	23.00	17.25	70.72	10.68	3.41	7.25
2.59	45.90	0.75	24.00	18.00	73.80	10.16	3.41	7.49
2.68	47.81	0.75	25.00	18.75	76.88	9.69	3.41	7.73
2.77	49.72	0.75	26.00	19.50	79.95	9.26	3.41	7.97
2.86	51.64	0.75	27.00	20.25	83.03	8.87	3.41	8.20
2.94	53.55	0.75	28.00	21.00	86.10	8.50	3.41	8.43
3.03	55.46	0.75	29.00	21.75	89.17	8.16	3.41	8.66
3.12	57.38	0.75	30.00	22.50	92.25	7.85	3.41	8.89
3.21	59.29	0.75	31.00	23.25	95.32	7.56	3.41	9.12
3.30	61.20	0.75	32.00	24.00	98.40	7.29	3.41	9.35
1.25	28.05	1.00	11.00	11.00	45.10	13.11	3.41	3.73
1.37	30.60	1.00	12.00	12.00	49.20	12.13	3.41	4.08
1.50	33.15	1.00	13.00	13.00	53.30	11.28	3.41	4.43
1.62	35.70	1.00	14.00	14.00	57.40	10.55	3.41	4.79
1.75	38.25	1.00	15.00	15.00	61.50	9.91	3.41	5.14

5.4. GEAR REDUCTION UNIT OF WOLKSVAGON AUTOMOBILE

The data of the gears used in WOLKSWAGON Automobile were measured from a dismounting gear-box. These measured data are tabulated in Table.5.15. The program was executed based on obtaining minimum volume of the pinion.

For the minimum volume case , input and output data of first gear, second gear, third gear, and fourth gear are shown in related tables respectively . Also, some acceptable output data for each cases are presented.

Table 5.15 The Dimensions of the Gears Used in WOLKSVAGON Automobile Gear-Box

Name	First Gear	Second Gear	Third Gear	Fourth Gear
Number of Teeth	34	30	26	22
Normal Module	2.5	2.5	2.5	2.5
Transverse Module	2.85	2.83	2.83	2.85
Pitch Diameter	96.60	84.90	73.58	62.70
Outside Diameter	98.70	84.70	71.40	61.20
Normal Press. Angle	20°	20°	20°	20°
Pitch Helix Angle	29°	28°	28°	29°
No of Teeth of Meshing gear	11	17	23	27
Center Distance	65	65	65	65
Velocity Ratio	3.090	1.764	1.130	0.814
Face width	11.30	11.30	15.40	17.80

Table 5.16. Input and Calculated Data of First Gear Pair Used in WOLKSVAGON
Gear-box for Single Stage Helical GRU

INPUT DATA	

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	62.00
The input speed value as (RPM).....:	3500.00
The output speed value as (RPM).....:	1132.00
Helix angle as (Degree).....:	28.67
Cycles of life.....:	100000
 OUTPUT OF THE PROGRAM	

The Smallest Volume (Cubic Cent.).....:	47.76
Module.....:	3.00
Face Width	34.47
Center Distance	85.93
Number of Teeth of Pinion.....:	14
Number of Teeth of Gear.....:	43
Pitch Diameter of Pinion	42.00
Pitch Diameter of Gear.....:	129.86
Gear Ratio.....:	3.09
Factor Of Safety.....:	3.26

Table 5.17. Input and Calculated Data of Second Gear Pair Used in WOLKSVAGON
Gear-box for Single Stage Helical GRU

INPUT DATA	

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	62.00
The input speed value as (RPM).....:	3000.00
The output speed value as (RPM).....:	1700.00
Helix angle as (Degree).....:	28.67
Cycles of life.....:	100000
 OUTPUT OF THE PROGRAM	

The Smallest Volume (Cubic Cent.).....:	55.72
Module.....:	3.50
Face Width	40.22
Center Distance	58.06
Number of Teeth of Pinion.....:	12
Number of Teeth of Gear.....:	21
Pitch Diameter of Pinion	42.00
Pitch Diameter of Gear.....:	74.12
Gear Ratio.....:	1.76
Factor Of Safety.....:	3.85

Table 5.18. Calculated Data of First Gear Pair Used in WOLKSWAGEN Gear-box for Single Step Helical GRU

VOLUME	CENT.D.	MODULE	NP	NG	DP	DG	FACE	S.FACT.
148.44	174.93	2.25	38	117	85.50	264.36	25.85	3.26
156.36	179.53	2.25	39	120	87.75	271.31	25.85	3.33
95.32	132.99	2.50	26	80	65.00	200.97	28.73	3.23
102.80	138.10	2.50	27	83	67.50	208.70	28.73	3.33
110.55	143.22	2.50	28	86	70.00	216.43	28.73	3.43
118.59	148.33	2.50	29	89	72.50	224.16	28.73	3.53
126.91	153.45	2.50	30	92	75.00	231.89	28.73	3.62
135.51	158.56	2.50	31	95	77.50	239.62	28.73	3.72
144.40	163.67	2.50	32	98	80.00	247.35	28.73	3.82
153.56	168.79	2.50	33	102	82.50	255.08	28.73	3.91
67.76	106.90	2.75	19	58	52.25	161.55	31.60	3.27
75.08	112.53	2.75	20	61	55.00	170.05	31.60	3.41
82.77	118.15	2.75	21	64	57.75	178.56	31.60	3.55
90.84	123.78	2.75	22	68	60.50	187.06	31.60	3.69
99.29	129.41	2.75	23	71	63.25	195.56	31.60	3.82
108.11	135.03	2.75	24	74	66.00	204.06	31.60	3.96
47.76	85.93	3.00	14	43	42.00	129.86	34.47	3.26
54.83	92.07	3.00	15	46	45.00	139.13	34.47	3.45
62.38	98.20	3.00	16	49	48.00	148.41	34.47	3.64
70.42	104.34	3.00	17	52	51.00	157.69	34.47	3.82
78.95	110.48	3.00	18	55	54.00	166.96	34.47	4.00
87.97	116.62	3.00	19	58	57.00	176.24	34.47	4.18
55.72	85.93	3.50	12	37	42.00	129.86	40.22	4.44
65.39	93.09	3.50	13	40	45.50	140.68	40.22	4.74
75.84	100.25	3.50	14	43	49.00	151.50	40.22	5.03
87.06	107.41	3.50	15	46	52.50	162.32	40.22	5.33
99.06	114.57	3.50	16	49	56.00	173.14	40.22	5.61
111.83	121.73	3.50	17	52	59.50	183.97	40.22	5.89
125.37	128.89	3.50	18	55	63.00	194.79	40.22	6.17
139.69	136.05	3.50	19	58	66.50	205.61	40.22	6.45
154.78	143.22	3.50	20	61	70.00	216.43	40.22	6.72
83.17	98.20	4.00	12	37	48.00	148.41	45.96	6.47
97.61	106.39	4.00	13	40	52.00	160.78	45.96	6.90
113.21	114.57	4.00	14	43	56.00	173.14	45.96	7.33
129.96	122.76	4.00	15	46	60.00	185.51	45.96	7.75
118.42	110.48	4.50	12	37	54.00	166.96	51.71	9.01
138.98	119.69	4.50	13	40	58.50	180.87	51.71	9.61
161.19	128.89	4.50	14	43	63.00	194.79	51.71	10.21
185.04	138.10	4.50	15	46	67.50	208.70	51.71	10.79
210.53	147.31	4.50	16	49	72.00	222.61	51.71	11.36
237.67	156.51	4.50	17	52	76.50	236.53	51.71	11.93
266.45	165.72	4.50	18	55	81.00	250.44	51.71	12.49
294.30	157.54	5.50	14	43	77.00	238.07	63.20	17.92
337.84	168.79	5.50	15	46	82.50	255.08	63.20	18.93
384.39	180.04	5.50	16	49	88.00	272.08	63.20	19.94
433.94	191.30	5.50	17	52	93.50	289.09	63.20	20.92
280.71	147.31	6.00	12	37	72.00	222.61	68.94	20.20
329.44	159.58	6.00	13	40	78.00	241.17	68.94	21.54

Table 5.19. Calculated Data of Second Gear Pair Used in WOLKSWAGEN Gear-box for Single Stage Helical GRU

VOLUME	CENT.D.	MODULE	NP	NG	DP	DG	FACE	S.FACT.
333.85	188.00	2.00	68	120	136.00	240.00	22.98	3.25
343.74	190.76	2.00	69	121	138.00	243.53	22.98	3.28
353.77	193.53	2.00	70	123	140.00	247.06	22.98	3.32
345.81	180.40	2.25	58	102	130.50	230.29	25.85	3.98
357.84	183.51	2.25	59	104	132.75	234.26	25.85	4.03
126.91	103.68	2.50	30	52	75.00	132.35	28.73	3.15
135.51	107.13	2.50	31	54	77.50	136.76	28.73	3.24
163.01	117.50	2.50	34	60	85.00	150.00	28.73	3.49
172.74	120.96	2.50	35	61	87.50	154.41	28.73	3.57
182.75	124.41	2.50	36	63	90.00	158.82	28.73	3.65
193.05	127.87	2.50	37	65	92.50	163.24	28.73	3.73
90.84	83.63	2.75	22	38	60.50	106.76	31.60	3.21
99.29	87.43	2.75	23	40	63.25	111.62	31.60	3.32
108.11	91.24	2.75	24	42	66.00	116.47	31.60	3.44
117.31	95.04	2.75	25	44	68.75	121.32	31.60	3.56
126.88	98.84	2.75	26	45	71.50	126.18	31.60	3.67
136.82	102.64	2.75	27	47	74.25	131.03	31.60	3.79
62.38	66.35	3.00	16	28	48.00	84.71	34.47	3.16
70.42	70.50	3.00	17	30	51.00	90.00	34.47	3.32
78.95	74.65	3.00	18	31	54.00	95.29	34.47	3.48
87.97	78.79	3.00	19	33	57.00	100.59	34.47	3.64
97.47	82.94	3.00	20	35	60.00	105.88	34.47	3.79
107.46	87.09	3.00	21	37	63.00	111.18	34.47	3.94
117.94	91.24	3.00	22	38	66.00	116.47	34.47	4.10
128.90	95.38	3.00	23	40	69.00	121.76	34.47	4.25
140.35	99.53	3.00	24	42	72.00	127.06	34.47	4.39
152.29	103.68	3.00	25	44	75.00	132.35	34.47	4.54
164.72	107.82	3.00	26	45	78.00	137.65	34.47	4.69
177.64	111.97	3.00	27	47	81.00	142.94	34.47	4.83
191.04	116.12	3.00	28	49	84.00	148.24	34.47	4.98
204.93	120.26	3.00	29	51	87.00	153.53	34.47	5.12
55.72	58.06	3.50	12	21	42.00	74.12	40.22	3.85
65.39	62.90	3.50	13	22	45.50	80.29	40.22	4.11
75.84	67.74	3.50	14	24	49.00	86.47	40.22	4.37
87.06	72.57	3.50	15	26	52.50	92.65	40.22	4.63
99.06	77.41	3.50	16	28	56.00	98.82	40.22	4.88
111.83	82.25	3.50	17	30	59.50	105.00	40.22	5.12
125.37	87.09	3.50	18	31	63.00	111.18	40.22	5.37
139.69	91.93	3.50	19	33	66.50	117.35	40.22	5.61
83.17	66.35	4.00	12	21	48.00	84.71	45.96	5.62
97.61	71.88	4.00	13	22	52.00	91.76	45.96	6.00
113.21	77.41	4.00	14	24	56.00	98.82	45.96	6.37
129.96	82.94	4.00	15	26	60.00	105.88	45.96	6.74
147.86	88.47	4.00	16	28	64.00	112.94	45.96	7.10
166.92	94.00	4.00	17	30	68.00	120.00	45.96	7.46
187.14	99.53	4.00	18	31	72.00	127.06	45.96	7.81
118.42	74.65	4.50	12	21	54.00	95.29	51.71	7.83
138.98	80.87	4.50	13	22	58.50	103.24	51.71	8.35
161.19	87.09	4.50	14	24	63.00	111.18	51.71	8.87

Table 5.20. Input and Calculated Data of Third Gear Pair Used in WOLKSWAGEN
Gear-box for Single Stage Helical GRU

INPUT DATA	

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	62.00
The input speed value as (RPM).....:	1132.00
The output speed value as (RPM).....:	1002.00
Helix angle as (Degree).....:	28.67
Cycles of life.....:	100000
 OUTPUT OF THE PROGRAM	

The Smallest Volume (Cubic Cent.).....:	187.14
Module.....:	4.00
Face Width	45.96
Center Distance	76.67
Number of Teeth of Pinion.....:	18
Number of Teeth of Gear.....:	20
Pitch Diameter of Pinion	72.00
Pitch Diameter of Gear.....:	81.34
Gear Ratio.....:	1.13
Factor Of Safety.....:	3.19

Table 5.21. Input and Calculated Data of Fourth Gear Pair Used in WOLKSWAGEN
Gear-box for Single Stage Helical GRU

INPUT DATA	

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).... :	1400.00
The input horse power..... :	62.00
The input speed value as (RPM).....:	1132.00
The output speed value as (RPM).....:	1390.00
Helix angle as (Degree).....:	28.67
Cycles of life.....:	100000
 OUTPUT OF THE PROGRAM	

The Smallest Volume (Cubic Cent.).....:	187.14
Module.....:	4.00
Face Width	45.96
Center Distance	80.20
Number of Teeth of Pinion.....:	18
Number of Teeth of Gear.....:	22
Pitch Diameter of Pinion	72.00
Pitch Diameter of Gear.....:	88.41
Gear Ratio.....:	1.23
Factor Of Safety.....:	3.19

Table 5.22. Calculated Data of Third Gear Pair Used in WOLKSWAGEN Gear-box for Single Stage Helical GRU

VOLUME	CENT.D.	MODULE	NP	NG	DP	DG	FACE	S.FACT.
817.57	193.27	2.75	66	74	181.50	205.05	31.60	3.23
842.53	196.20	2.75	67	75	184.25	208.15	31.60	3.27
867.87	199.13	2.75	68	76	187.00	211.26	31.60	3.31
561.42	153.34	3.00	48	54	144.00	162.68	34.47	3.18
585.05	156.54	3.00	49	55	147.00	166.07	34.47	3.23
609.18	159.73	3.00	50	56	150.00	169.46	34.47	3.29
633.79	162.93	3.00	51	57	153.00	172.85	34.47	3.34
658.89	166.12	3.00	52	58	156.00	176.24	34.47	3.40
303.36	104.36	3.50	28	31	98.00	110.71	40.22	3.16
371.85	115.54	3.50	31	35	108.50	122.58	40.22	3.43
396.23	119.27	3.50	32	36	112.00	126.53	40.22	3.52
421.38	122.99	3.50	33	37	115.50	130.49	40.22	3.61
447.30	126.72	3.50	34	38	119.00	134.44	40.22	3.70
474.00	130.45	3.50	35	39	122.50	138.39	40.22	3.79
187.14	76.67	4.00	18	20	72.00	81.34	45.96	3.19
208.51	80.93	4.00	19	21	76.00	85.86	45.96	3.34
231.04	85.19	4.00	20	22	80.00	90.38	45.96	3.48
254.72	89.45	4.00	21	23	84.00	94.90	45.96	3.63
279.55	93.71	4.00	22	24	88.00	99.42	45.96	3.77
305.54	97.97	4.00	23	25	92.00	103.94	45.96	3.91
332.69	102.23	4.00	24	27	96.00	108.46	45.96	4.05
360.99	106.49	4.00	25	28	100.00	112.97	45.96	4.19
390.45	110.75	4.00	26	29	104.00	117.49	45.96	4.33
421.06	115.01	4.00	27	30	108.00	122.01	45.96	4.47
452.83	119.27	4.00	28	31	112.00	126.53	45.96	4.60
485.75	123.52	4.00	29	32	116.00	131.05	45.96	4.74
519.83	127.78	4.00	30	33	120.00	135.57	45.96	4.87
555.06	132.04	4.00	31	35	124.00	140.09	45.96	5.00
591.45	136.30	4.00	32	36	128.00	144.61	45.96	5.14
629.00	140.56	4.00	33	37	132.00	149.13	45.96	5.27
667.69	144.82	4.00	34	38	136.00	153.64	45.96	5.40
707.55	149.08	4.00	35	39	140.00	158.16	45.96	5.53
748.56	153.34	4.00	36	40	144.00	162.68	45.96	5.65
790.72	157.60	4.00	37	41	148.00	167.20	45.96	5.78
834.04	161.86	4.00	38	42	152.00	171.72	45.96	5.91
878.51	166.12	4.00	39	44	156.00	176.24	45.96	6.04
924.14	170.38	4.00	40	45	160.00	180.76	45.96	6.16
970.93	174.64	4.00	41	46	164.00	185.28	45.96	6.29
210.53	76.67	4.50	16	18	72.00	81.34	51.71	4.04
237.67	81.46	4.50	17	19	76.50	86.43	51.71	4.25
266.45	86.25	4.50	18	20	81.00	91.51	51.71	4.45
296.88	91.05	4.50	19	21	85.50	96.59	51.71	4.66
328.96	95.84	4.50	20	22	90.00	101.68	51.71	4.86
362.67	100.63	4.50	21	23	94.50	106.76	51.71	5.06
398.04	105.42	4.50	22	24	99.00	111.84	51.71	5.26
435.04	110.21	4.50	23	25	103.50	116.93	51.71	5.46
473.70	115.01	4.50	24	27	108.00	122.01	51.71	5.65
513.99	119.80	4.50	25	28	112.50	127.10	51.71	5.85
555.93	124.59	4.50	26	29	117.00	132.18	51.71	6.04
599.52	129.38	4.50	27	30	121.50	137.26	51.71	6.23
644.75	134.17	4.50	28	31	126.00	142.35	51.71	6.42
691.63	138.97	4.50	29	32	130.50	147.43	51.71	6.60
740.15	143.76	4.50	30	33	135.00	152.51	51.71	6.79
790.32	148.55	4.50	31	35	139.50	157.60	51.71	6.97

**Table 5.23. Calculated Data of Fourth Gear Pair Used in WOLKSWAGEN Gear-box
for Single Stage Helical GRU**

VOLUME	CENT. D.	MODULE	NP	NG	DP	DG	FACE	S. FACT.
371.85	120.86	3.50	31	38	108.50	133.23	40.22	3.43
396.23	124.76	3.50	32	39	112.00	137.53	40.22	3.52
421.38	128.66	3.50	33	40	115.50	141.82	40.22	3.61
447.30	132.56	3.50	34	41	119.00	146.12	40.22	3.70
474.00	136.46	3.50	35	42	122.50	150.42	40.22	3.79
501.47	140.36	3.50	36	44	126.00	154.72	40.22	3.88
529.72	144.26	3.50	37	45	129.50	159.02	40.22	3.97
558.74	148.16	3.50	38	46	133.00	163.31	40.22	4.06
588.54	152.06	3.50	39	47	136.50	167.61	40.22	4.14
619.10	155.95	3.50	40	49	140.00	171.91	40.22	4.23
650.45	159.85	3.50	41	50	143.50	176.21	40.22	4.32
682.56	163.75	3.50	42	51	147.00	180.50	40.22	4.40
187.14	80.20	4.00	18	22	72.00	88.41	45.96	3.19
208.51	84.66	4.00	19	23	76.00	93.32	45.96	3.34
231.04	89.12	4.00	20	24	80.00	98.23	45.96	3.48
254.72	93.57	4.00	21	25	84.00	103.14	45.96	3.63
210.53	80.20	4.50	16	19	72.00	88.41	51.71	4.04
237.67	85.22	4.50	17	20	76.50	93.94	51.71	4.25
266.45	90.23	4.50	18	22	81.00	99.46	51.71	4.45
296.88	95.24	4.50	19	23	85.50	104.99	51.71	4.66
328.96	100.26	4.50	20	24	90.00	110.51	51.71	4.86
362.67	105.27	4.50	21	25	94.50	116.04	51.71	5.06
398.04	110.28	4.50	22	27	99.00	121.56	51.71	5.26
435.04	115.29	4.50	23	28	103.50	127.09	51.71	5.46
473.70	120.31	4.50	24	29	108.00	132.61	51.71	5.65
513.99	125.32	4.50	25	30	112.50	138.14	51.71	5.85
555.93	130.33	4.50	26	31	117.00	143.67	51.71	6.04
599.52	135.35	4.50	27	33	121.50	149.19	51.71	6.23
644.75	140.36	4.50	28	34	126.00	154.72	51.71	6.42
691.63	145.37	4.50	29	35	130.50	160.24	51.71	6.60
740.15	150.38	4.50	30	36	135.00	165.77	51.71	6.79
790.32	155.40	4.50	31	38	139.50	171.29	51.71	6.97
842.13	160.41	4.50	32	39	144.00	176.82	51.71	7.16
895.58	165.42	4.50	33	40	148.50	182.35	51.71	7.34
950.68	170.44	4.50	34	41	153.00	187.87	51.71	7.52
288.79	89.12	5.00	16	19	80.00	98.23	57.45	5.44
326.02	94.69	5.00	17	20	85.00	104.37	57.45	5.72
365.51	100.26	5.00	18	22	90.00	110.51	57.45	6.00
407.25	105.83	5.00	19	23	95.00	116.65	57.45	6.28
451.24	111.40	5.00	20	24	100.00	122.79	57.45	6.55
497.49	116.97	5.00	21	25	105.00	128.93	57.45	6.82
546.00	122.54	5.00	22	27	110.00	135.07	57.45	7.08
596.77	128.11	5.00	23	28	115.00	141.21	57.45	7.35
649.79	133.67	5.00	24	29	120.00	147.35	57.45	7.61
705.07	139.24	5.00	25	30	125.00	153.49	57.45	7.87
762.60	144.81	5.00	26	31	130.00	159.63	57.45	8.13
822.39	150.38	5.00	27	33	135.00	165.77	57.45	8.38

5.5. DOUBLE STAGE GEAR REDUCTION UNIT

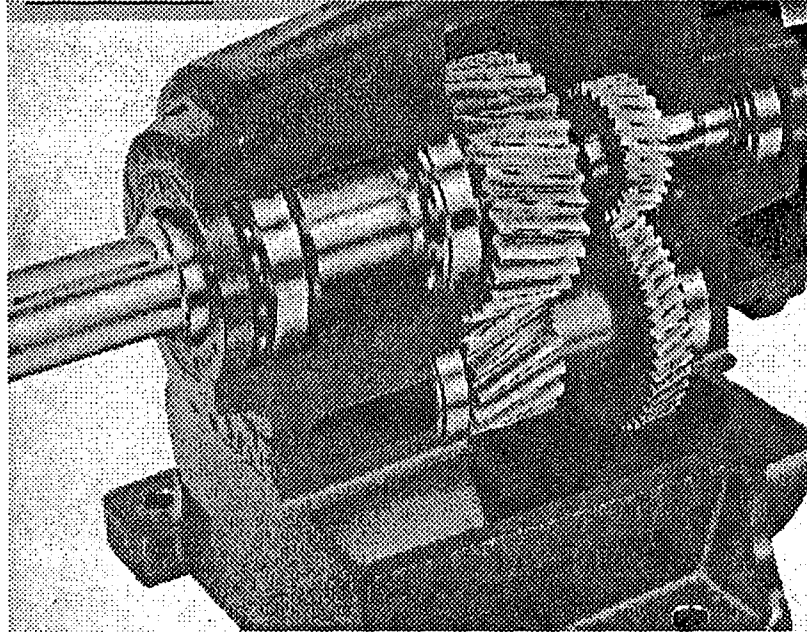


Figure 5.2 Double Stage Helical Gear Reduction Unit [43]

5.5.1 Helical Gear Reduction Unit of Yılmaz Redüktör

Table 5.24 Original Data for Helical GRU of Yılmaz Redüktör
($a_w=95.5$ Reference: MY085/98)

	First Stage				Second Stage			
#	m	N _p	N _g	H.Ang.	M	N _p	N _g	H.Ang.
1	2.00	12	76	20°	3.25	15	40	15°
2	2.00	28	60	20°	3.25	18	37	15°

EXAMPLE I:

Table 5.25.: Input and Output Data for Double Stage
Helical GRU of YILMAZ REDÜKTÖR

INPUT DATA FOR FIRST STAGE

Ultimate tensile strength. (MPa).....:	744.82
The input horse power.....:	13.40
The input speed value as (RPM).....:	3000.00
The output speed value as (RPM).....:	940.16
Helix angle as (Degree).....:	20.00
Cycles of life.....:	1000000
The type of the material.....:	STEEL

OUTPUT OF THE PROGRAM FOR FIRST STAGE

The Smallest Volume (Cubic Cent.).....:	25.62
Module.....:	2.00
Face Width	36.74
Center Distance	62.44
Number of Teeth of Pinion.....:	14
Number of Teeth of Gear.....:	44
Pitch Diameter of Pinion	29.80
Pitch Diameter of Gear.....:	95.08
Gear Ratio.....:	3.19
Factor Of Safety.....:	4.21

INPUT DATA FOR SECOND STAGE

The type of the material.....:	STEEL
Ultimate tensile strength. (MPa).....:	744.82
The input horse power.....:	13.40
The input speed value as (RPM).....:	940.16
The output speed value as (RPM).....:	524.06
Helix angle as (Degree).....:	20.00
Cycles of life.....:	1000000

OUTPUT OF THE PROGRAM FOR SECOND STAGE

The Smallest Volume (Cubic Cent.).....:	86.47
Module.....:	3.00
Face Width	55.11
Center Distance	62.44
Number of Teeth of Pinion.....:	14
Number of Teeth of Gear.....:	25
Pitch Diameter of Pinion	44.70
Pitch Diameter of Gear.....:	80.18
Gear Ratio.....:	1.79
Factor Of Safety.....:	4.48

EXAMPLE II:

Table 5.26: Input and Output Data for Double Stage
Helical GRU of YILMAZ REDÜKTÖR

INPUT DATA FOR FIRST STAGE

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).....:	744.82
The input horse power.....:	13.40
The input speed value as (RPM).....:	4000.00
The output speed value as (RPM).....:	785.27
Helix angle as (Degree).....:	20.00
Cycles of life.....:	1000000

OUTPUT OF THE PROGRAM FOR FIRST STAGE

The Smallest Volume (Cubic Cent.).....:	25.62
Module.....:	2.00
Face Width	36.74
Center Distance	90.79
Number of Teeth of Pinion.....:	14
Number of Teeth of Gear.....:	71
Pitch Diameter of Pinion	29.80
Pitch Diameter of Gear.....:	151.78
Gear Ratio.....:	5.09
Factor Of Safety.....:	5.49

INPUT DATA FOR SECOND STAGE

The type of the material.....:	STEEL
Ultimate tensile strength.(MPa).....:	744.82
The input horse power.....:	13.40
The input speed value as (RPM).....:	785.27
The output speed value as (RPM).....:	307.40
Helix angle as (Degree).....:	20.00
Cycles of life.....:	1000000

OUTPUT OF THE PROGRAM FOR SECOND STAGE

The Smallest Volume (Cubic Cent.).....:	112.94
Module.....:	3.00
Face Width	55.11
Center Distance	90.78
Number of Teeth of Pinion.....:	16
Number of Teeth of Gear.....:	40
Pitch Diameter of Pinion	51.08
Pitch Diameter of Gear.....:	130.49
Gear Ratio.....:	2.55
Factor Of Safety.....:	4.23

5.5.2 Spur Gear Reduction Unit:

EXAMPLE :

Table 5.27: Input and Output Data for Double Stage
Spur GRU

INPUT DATA FOR FIRST STAGE

TYPE OF THE MATERIAL.....:	Steel
ULTIMATE TENSILE STRENGTH (MPa).....:	744.82
THE BRINELL HARDNESS NUMBER	220.00
THE INPUT POWER(HORSE POWER).....:	30.00
THE INPUT SPEED VALUE (RPM).....:	3000.00
THE GEAR RATIO.....:	2.87

OUTPUT OF THE PROGRAM FOR FIRST STAGE

THE SMALLEST VOLUME (Cubic Centimeter).....:	292.04
MODULE.....:	1.50
THE FACE WIDTH	21.34
CENTER DISTANCE.....:	255.42
WHOLE DEPTH OF CUT	3.25
THE NUMBER OF PINION TEETH	88
THE NUMBER OF GEAR TEETH	252
PINION PITCH DIAMETER	132.00
GEAR PITCH DIAMETER	378.84
PINION OUTSIDE DIAMETER	135.00
GEAR OUTSIDE DIAMETER	381.84
FACTOR OF SAFETY FOR BENDING.....:	3.41
FACTOR OF SAFETY FOR SURFACE DURAB.....:	3.44

INPUT DATA FOR SECOND STAGE

TYPE OF THE MATERIAL.....:	Steel
ULTIMATE TENSILE STRENGTH	744.82
THE BRINELL HARDNESS NUMBER	220.00
THE INPUT POWER(HORSE POWER).....:	30.00
THE INPUT SPEED VALUE (RPM).....:	1045.30
THE GEAR RATIO.....:	1.39

OUTPUT OF THE PROGRAM FOR SECOND STAGE

THE SMALLEST VOLUME (Cubic Centimeter).....:	1003.86
MODULE.....:	2.00
THE FACE WIDTH	27.91
CENTER DISTANCE.....:	256.13
WHOLE DEPTH OF CUT	4.33
THE NUMBER OF PINION TEETH	107
THE NUMBER OF GEAR TEETH	149
PINION PITCH DIAMETER	214.00
GEAR PITCH DIAMETER	298.26
PINION OUTSIDE DIAMETER	218.00
GEAR OUTSIDE DIAMETER	302.26
FACTOR OF SAFETY FOR BENDING.....:	3.41
FACTOR OF SAFETY FOR SURFACE DURAB.....:	3.44

CHAPTER 6

DISCUSSION AND CONCLUSION

6.1 INTRODUCTION

The work, reported in the previous chapters, has covered the related subjects, and methods used to design the gear reduction unit. The methods and theories on the design of spur and helical gear pairs are reviewed and design procedures are developed for computer program.

6.2 DISCUSSIONS ON THE PROGRAM

The program automates the design of both spur and helical gear pair in single and double stage gear reduction unit subjected to the design limitations. During the running of the computer program, AGMA geometric parameters (J , I) are calculated by making interpolation. Running of the program is simplified with some explanations for each input value , so that anyone who knows little about the Gear Reduction Unit, can use the program without any difficulty. The program writes all the possible solutions in a data file and gives the optimum value together with input data on the screen.

The optimisation technique used is basically an iteration method (linear search method) and runs all the practical values of standard module and pinion teeth number through their possible range in an order tracing all possible combinations and looking for a solution set with minimum volume occupied by the pinion. Program's structures

were prepared implicitly in order to modify it for obtaining any other minimum parameter when desired. Volume of the gears (pinion) at design load is selected as the objective function for the optimisation. By this way, it is possible to obtain the minimum inertia of the system by inserting some formulas (as objective function) into the program after getting optimum parameters of the gears.

6.3 DISCUSSION OF RESULTS

6.3.1 The Gear-box of a MURAT-131 Automobile

The first example is the gear-box of a MURAT-131 Automobile. The mechanical properties of the material used for the gears are given by TOFAS as 1200-1600 MPa for ultimate tensile strength and 950 MPa or higher for yield strength. Since there is no detail information about the material, ultimate strength is taken as 1400 MPa. During strength calculations endurance strength is assumed as equal to half of ultimate strength.

In order to have an idea about the tested model, the characteristics of the gears are given in table.5.1. The assembly of the gear-box is also given in figure.5.1. As seen from table.5.1 the gear pitch diameters of the first (constant meshing), second, third and fourth gears are 81.660, 94.327, 77.1764 and 61.248 mm. Respectively. The calculated values are 64.15, 115.49, 66.17 and 61.71 mm respectively. These values are optimum for each separate gear pairs. Calculated pitch diameters are smaller than the original values. This difference is the result of taking slightly difference input values(mounting conditions, unknown parameters etc.).

The pressure and helix angles are assumed as 20° and 30° respectively, whereas in the model, the pressure angle of 14.5° is used for the constant meshing, third and fourth gear, and 20° is used for the second gear. Also 31° is used as a helix angle for constant meshing and fourth gear, and 29° is used for the second and third gear.

The normal module at the conditions of the minimum volume was calculated as 2.75, 3.50, 3.50, and 3.00 for first, second, third and fourth gears respectively. The original values are taken as 2.50.

The number of teeth are calculated as 23, 32, 18, and 20 for the gears respectively. The original values are tabulated as 28, 33, 27, 21 in Table 5.1.

6.3.2 Gear Reduction Unit of Black and Decker Hand Drill

In the gear reduction unit of the Black and Decker Hammer hand drill two pairs of spur gears are used. The original parameters for the gear reduction unit were taken from a dismantled gearbox. These parameters are tabulated in table 5.27. The mechanical properties of the material used for the gears are calculated by measuring the hardness number. At the teeth surface, the hardness was measured as 200 HB. For steels, this number is corresponding to 600 MPa.

As seen from Table 5.10 the gear pitch diameters of the first and second gears are 24.75, 30.75 mm respectively. The calculated values are 26.08 and 45.10 mm respectively.. These values are greater than the original value. This difference is due to using non-standard gears.

6.3.3 The Gear-box of a WOLKSVAGON Automobile

The original parameters for the gear-box of a Wolkswagen automobile were taken from a dismantled gearbox. Those parameters are tabulated in table 5.15. For the calculation, ultimate tensile strength of the gear material was taken as 1400 MPa. as in the case of Murat-131 gear-box. During strength calculations endurance strength is assumed as equal to half of ultimate tensile strength.

As seen from table 5.15 the gear pitch diameters of the first, second, third and fourth gears are 96.60, 84.90, 73.58 and 62.70 mm respectively. The calculated values are 129.86, 74.12, 81.34 and 88.41 mm respectively. These values are optimum for each separate gear pairs. Calculated pitch diameter of three gear pairs are greater than the original values. But, they have smaller face width.

The pressure angles are assumed as 20° and helix angles are measured as 28° for the second and third gears and 29° for the first and forth gears.

The normal module at the conditions of the minimum volume were calculated as 3.00, 3.50, 4.00 and 4.00 for first, second, third and fourth gears respectively. The original values were measured as 2.50.

The number of gear teeth are calculated as 43, 21, 20, and 22 for the gears respectively. The original values were 34, 30, 26, 22.

6.3.4. Double Stage Helical GRU of 'YILMAZ REDÜKTÖR'

As indicated in Ref.[44], 21NiCrMo2 material was used for gear material. This material has ultimate tensile strength and yield strength as 744.82 MPa. and 472 MPa. respectively.

In the original data the gear teeth for the first and second stage are taken as 12/76, and 15/40 respectively. The center distance is taken as 92.52 mm for the double stage helical GRU in Ref.[44].

With the program the gear teeth for first and second stage are found as 14/71, and 16/40 as shown in Table 5.26 .The center distance is found as 90.78 mm. which is smaller than the given value.

6.4 CONCLUSIONS

The program developed in this thesis presents a design procedure for an accurate, complete and fast design method for gear-transmission systems. This program provides all possible solutions as to satisfy the limitations on center distance for a given input data. The output data can be limited by putting some limitations as control statements in the program. This reduces the number of solutions to reasonable alternatives. In the case of helical gears, the variation of the center distance can be regulated by small variations in the helix angle easily. The solutions can be arranged in an order in case of volume, face width, center distance etc... for the design of the gear reduction unit, if there is any space limitation.

The outputs of the program show that the most important parameters are module and number of teeth. Volume and center distance of the gears decreases when number of teeth decrease for the same module. The obtained results show that both minimum volume and center distance are obtained at same stage for the common input values. Program is valid for different types of applications.

A systematic approach is developed for the design and optimization of spur and helical gear pairs. The methods and theories used in this work, and the results of the computer program are verified with practical examples for Murat-131, Wolkswagon automobiles and a hand drill (Black & Decker) and reasonable solutions were obtained for single stage gear reduction units. Few examples were presented for double stage gear reduction units.

With the computer package program developed, it would naturally be possible to reduce the time of design and optimization of single and double stage gear reduction unit which is one of the main aim of CAD applications.

6.5. SUGGESTIONS FOR FUTURE WORK

This work, on the computer aided design and optimisation of gear reduction unit with minimum volume can be extended to the following fields to provide options to the machine tool designer.

- a) It can be modified for the multistage gear reduction unit.
- b) It should be integrated with other design and/or selection of other elements such as shafts, bearings, keys and sealings.
- c) Dynamic performance of the gear reduction unit can be tested. The stiffness', inertias and frictions are important parameters which can be calculated after sizing of the parameters of gear reduction unit.



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APPENDICES



APPENDIX-A

NECESSARY TABLES

TABLE A.1. AGMA GEOMETRY FACTOR J FOR TEETH HAVING $\phi = 20^\circ$,
 $a = 1.000$ in, $b = 1.250$ in, and $r_f = 0.300$ in

Number of teeth	Number of teeth in mating gear						
	17	25	35	50	85	300	1000
18	0.324 04	0.332 14	0.338 40	0.344 04	0.350 50	0.355 94	0.361 12
19	0.330 29	0.338 78	0.345 37	0.351 34	0.358 22	0.364 05	0.369 63
20	0.336 00	0.344 85	0.351 76	0.358 04	0.365 32	0.371 51	0.377 49
21	0.341 24	0.350 44	0.357 64	0.364 22	0.371 86	0.378 41	0.384 75
22	0.346 07	0.355 59	0.363 06	0.369 92	0.377 92	0.384 79	0.391 48
24	0.354 68	0.364 77	0.372 75	0.380 12	0.388 77	0.396 26	0.403 60
26	0.362 11	0.372 72	0.381 15	0.388 57	0.398 21	0.406 25	0.414 18
28	0.368 60	0.379 67	0.388 51	0.396 73	0.406 50	0.415 04	0.423 51
30	0.374 62	0.385 80	0.395 00	0.403 59	0.413 83	0.422 83	0.431 75
34	0.383 94	0.396 11	0.405 94	0.415 17	0.426 24	0.436 04	0.445 86
38	0.391 70	0.404 46	0.414 80	0.424 56	0.436 33	0.446 80	0.457 35
45	0.402 23	0.415 79	0.426 85	0.437 35	0.450 16	0.461 52	0.473 10
50	0.408 08	0.422 08	0.433 55	0.444 48	0.457 78	0.469 75	0.481 93
60	0.417 02	0.431 73	0.443 83	0.455 42	0.469 60	0.482 43	0.495 57
75	0.426 20	0.441 63	0.454 40	0.466 68	0.481 79	0.495 54	0.509 70
100	0.435 61	0.451 80	0.465 27	0.478 27	0.494 37	0.509 09	0.524 35
150	0.445 30	0.462 26	0.476 45	0.490 23	0.507 36	0.523 12	0.539 54
300	0.455 26	0.473 04	0.487 98	0.502 56	0.520 78	0.537 65	0.555 33
Rack	0.465 54	0.484 15	0.499 88	0.515 29	0.534 67	0.552 72	0.571 73

TABLE A.2. AGMA GEOMETRY FACTOR J FOR TEETH HAVING $\phi = 20^\circ$,
 $a = 0.800$ in, $b = 1.000$ in, AND $r_f = 0.304$ in

Number of teeth	Number of teeth in mating gear						
	17	25	35	50	85	300	1000
14	0.321 54	0.328 92	0.334 52	0.339 46	0.345 07	0.349 70	0.354 05
15	0.328 49	0.336 24	0.342 14	0.347 40	0.353 36	0.358 31	0.362 98
16	0.334 67	0.342 76	0.348 94	0.354 47	0.360 75	0.365 99	0.370 96
17	0.340 21	0.348 60	0.355 04	0.360 81	0.367 39	0.372 91	0.378 15
18	0.345 20	0.353 87	0.360 54	0.366 54	0.373 40	0.379 16	0.384 66
19	0.349 72	0.358 64	0.365 53	0.371 74	0.378 85	0.384 85	0.390 58
20	0.354 83	0.362 99	0.370 07	0.376 47	0.383 82	0.390 03	0.395 99
21	0.357 59	0.366 96	0.374 23	0.380 81	0.388 38	0.394 79	0.400 95
22	0.361 04	0.370 61	0.378 05	0.384 79	0.392 56	0.399 16	0.405 51
24	0.367 15	0.377 08	0.384 82	0.391 86	0.400 00	0.406 94	0.413 63
26	0.372 40	0.382 64	0.390 64	0.397 94	0.406 41	0.413 64	0.420 64
28	0.376 90	0.387 47	0.395 71	0.403 24	0.411 95	0.419 48	0.426 76
30	0.380 96	0.391 71	0.400 15	0.407 88	0.416 89	0.424 62	0.432 13
34	0.387 64	0.398 80	0.407 59	0.415 66	0.425 10	0.433 23	0.441 15
38	0.393 01	0.404 49	0.413 56	0.421 91	0.431 71	0.440 16	0.448 43
45	0.400 23	0.412 15	0.421 61	0.430 34	0.440 62	0.449 52	0.458 26
50	0.404 21	0.416 38	0.426 05	0.435 00	0.445 55	0.454 70	0.463 70
60	0.410 28	0.422 82	0.432 81	0.442 09	0.453 06	0.462 60	0.472 01
75	0.416 45	0.429 38	0.439 71	0.449 32	0.460 72	0.470 67	0.480 51
100	0.422 74	0.436 06	0.446 74	0.456 70	0.468 54	0.478 91	0.489 19
150	0.429 16	0.442 87	0.453 91	0.464 22	0.476 53	0.487 33	0.498 07
300	0.435 71	0.449 83	0.461 23	0.471 91	0.484 69	0.495 94	0.507 16
Rack	0.442 40	0.456 94	0.468 70	0.479 77	0.493 03	0.504 76	0.516 47

TABLE A.3. AGMA GEOMETRY FACTOR J FOR TEETH HAVING $\phi = 25^\circ$,
 $a = 1.000$ in, $b = 1.250$ in, AND $r_f = 0.300$ in

Number of teeth	Number of teeth in mating gear						
	17	25	35	50	85	300	1000
13	0.346 84	0.352 92	0.357 14	0.361 78	0.365 72	0.369 25	0.372 51
14	0.359 24	0.365 87	0.370 81	0.375 14	0.379 94	0.383 86	0.387 49
15	0.370 27	0.377 40	0.382 75	0.387 44	0.392 67	0.396 94	0.400 92
16	0.380 16	0.387 75	0.393 46	0.398 49	0.404 11	0.408 73	0.413 03
17	0.389 07	0.397 09	0.403 14	0.408 49	0.414 48	0.419 41	0.424 02
18	0.397 14	0.405 56	0.411 93	0.417 56	0.423 90	0.429 13	0.434 03
19	0.404 49	0.413 28	0.419 94	0.425 85	0.432 50	0.438 01	0.443 18
20	0.411 21	0.420 34	0.427 27	0.433 44	0.440 39	0.446 16	0.451 59
21	0.417 38	0.426 82	0.434 01	0.440 42	0.447 65	0.453 67	0.459 33
22	0.423 06	0.432 80	0.440 23	0.446 86	0.454 36	0.460 60	0.466 50
24	0.433 18	0.443 46	0.451 32	0.458 35	0.466 35	0.473 01	0.479 32
26	0.441 93	0.452 68	0.460 93	0.468 33	0.476 74	0.483 78	0.490 46
28	0.449 57	0.460 75	0.469 33	0.477 05	0.485 85	0.493 23	0.500 23
30	0.456 31	0.467 85	0.476 75	0.484 75	0.493 89	0.501 57	0.508 68
34	0.467 63	0.479 81	0.489 23	0.497 72	0.507 46	0.515 66	0.523 49
38	0.476 78	0.489 48	0.499 33	0.508 24	0.518 47	0.527 10	0.535 36
45	0.489 19	0.502 61	0.513 05	0.522 52	0.533 44	0.542 68	0.551 54
50	0.496 08	0.509 91	0.520 68	0.530 47	0.541 77	0.551 36	0.560 56
60	0.506 83	0.521 09	0.532 38	0.542 67	0.554 57	0.564 69	0.574 44
75	0.517 47	0.532 57	0.544 40	0.555 20	0.567 73	0.578 42	0.588 73
100	0.528 60	0.544 36	0.556 76	0.568 10	0.581 29	0.592 57	0.603 48
150	0.540 05	0.556 51	0.569 51	0.581 58	0.595 26	0.607 16	0.618 69
300	0.551 85	0.569 02	0.582 59	0.595 57	0.609 67	0.622 22	0.634 42
Rack	0.564 05	0.581 94	0.596 13	0.609 21	0.624 56	0.637 78	0.650 68

TABLE A.4. AGMA GEOMETRY FACTOR J FOR TEETH HAVING $\phi = 25^\circ$,
 $a = 1.000$ in, $b = 1.350$ in, AND $r_f = 0.245$ in

Number of teeth	Number of teeth in mating gear						
	17	25	35	50	85	300	1000
14	0.333 63	0.339 13	0.343 18	0.346 71	0.350 62	0.353 79	0.356 73
15	0.345 19	0.351 08	0.355 47	0.359 32	0.363 59	0.367 07	0.370 30
16	0.355 52	0.361 82	0.366 53	0.370 67	0.375 27	0.379 04	0.382 55
17	0.364 84	0.371 51	0.376 52	0.380 93	0.385 85	0.389 89	0.393 66
18	0.373 29	0.380 31	0.385 60	0.390 26	0.395 48	0.399 77	0.403 78
19	0.380 99	0.388 13	0.393 88	0.398 79	0.404 27	0.408 80	0.413 04
20	0.388 03	0.395 68	0.401 46	0.406 59	0.412 34	0.417 09	0.421 55
21	0.394 50	0.402 43	0.408 44	0.413 77	0.419 77	0.424 73	0.429 39
22	0.400 47	0.408 66	0.414 88	0.420 40	0.426 63	0.431 79	0.436 64
24	0.411 11	0.419 78	0.426 38	0.432 25	0.438 90	0.444 42	0.449 63
26	0.420 31	0.429 40	0.436 34	0.442 53	0.449 55	0.455 39	0.460 91
28	0.428 35	0.437 82	0.445 06	0.451 53	0.458 88	0.465 01	0.470 81
30	0.435 44	0.445 24	0.452 75	0.459 48	0.467 12	0.473 51	0.479 57
34	0.447 38	0.457 75	0.465 72	0.472 87	0.481 03	0.487 87	0.494 37
38	0.457 04	0.467 88	0.476 22	0.483 74	0.492 32	0.499 54	0.506 40
45	0.470 16	0.481 63	0.490 51	0.498 51	0.507 69	0.515 42	0.522 60
50	0.477 45	0.489 29	0.498 45	0.506 74	0.516 25	0.524 28	0.531 95
60	0.488 63	0.501 03	0.510 65	0.519 37	0.529 40	0.537 89	0.546 02
75	0.500 13	0.513 10	0.523 20	0.532 37	0.542 94	0.551 91	0.560 52
100	0.511 97	0.525 53	0.536 12	0.545 75	0.556 90	0.566 37	0.575 49
150	0.524 18	0.538 35	0.549 44	0.559 56	0.571 30	0.581 29	0.590 94
300	0.536 61	0.551 60	0.563 21	0.573 83	0.586 18	0.596 72	0.606 92
Rack	0.549 90	0.565 32	0.577 47	0.588 60	0.601 58	0.612 69	0.623 46

Table A5: Size Factors for Spur Gear Teeth

Module m	Size Factors	Module m	Size Factors
1 to 2	1.000	11	0.843
2.25	0.984	12	0.836
2.50	0.974	14	0.824
2.75	0.965	16	0.813
3.00	0.956	18	0.804
3.50	0.942	20	0.796
4.00	0.930	22	0.788
4.50	0.920	25	0.779
5.00	0.910	28	0.770
5.50	0.902	32	0.760
6.00	0.894	36	0.752
7.00	0.881	40	0.744
8.00	0.870	45	0.736
9.00	0.860	50	0.728
10.0	0.851		

Table A6: Load Distribution Factor K_m

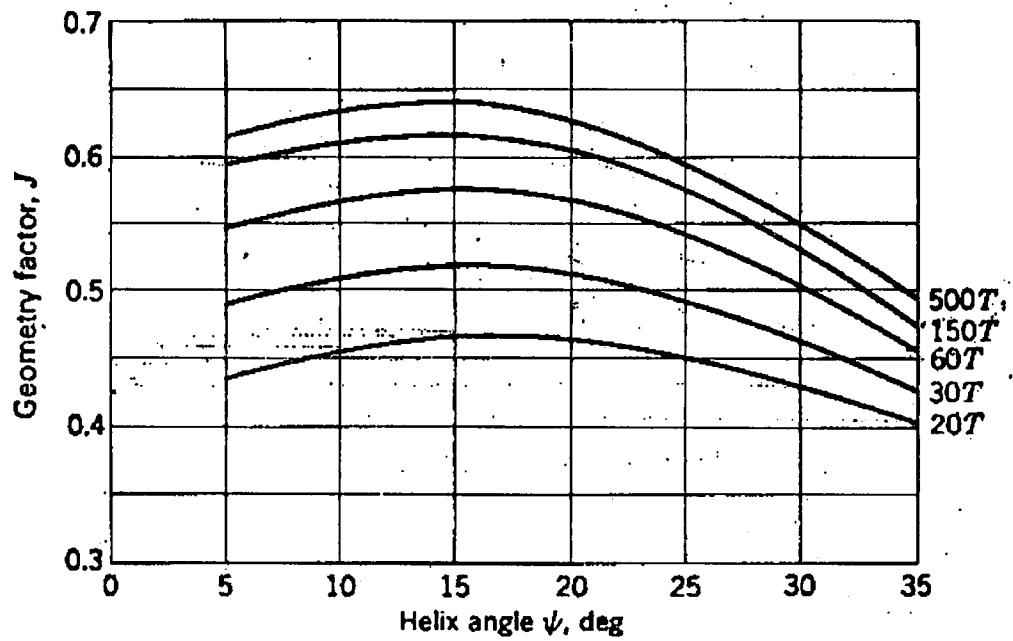
Characteristics of support	Face Width, mm			
	0 to 50	150	225	400 up
Accurate mountings, small bearing clearance, minimum deflection, precision	1.3	1.4	1.5	1.8
Less rigid mountings, less accurate gears, contact across full face	1.6	1.7	1.8	2.2
Accuracy and mounting such that less Than full-face contact exists	Over 2.2			

Table A7: Values of the Elastic Coefficient (C_p) for Spur and Helical Gears with Nonlocalized Contact and for $\nu = 0.30$

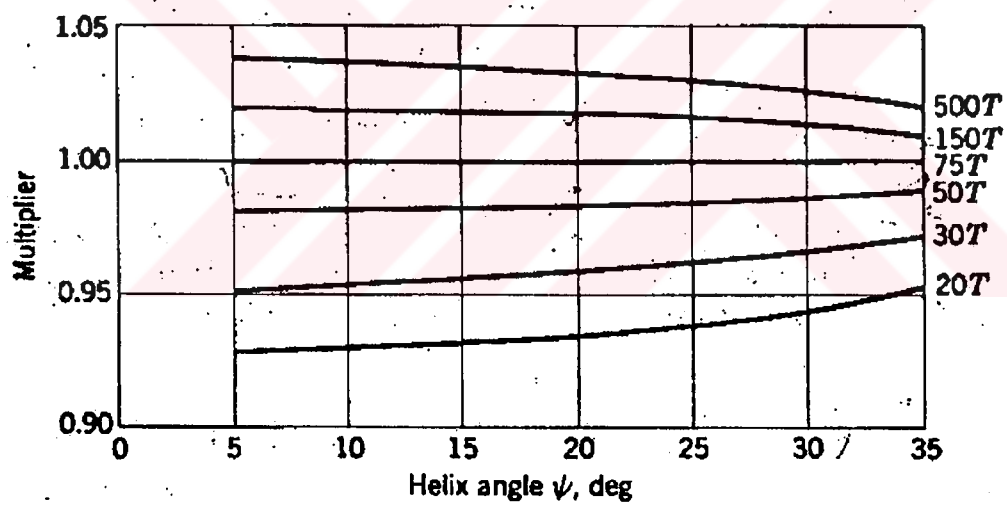
Pinion	Gear						
	E, GPa	Steel	Malleab. Iron	Nodular iron	Cast iron	Alumin. Bronze	Tin bronze
Steel	200	191	181	179	174	162	158
Malleab. Iron	170	181	174	172	168	158	154
Nodular iron	170	179	172	170	166	156	152
Cast iron	150	174	168	166	163	154	149
Al. Bronze	120	162	158	156	154	145	141
Tin bronze	110	158	154	152	149	141	137

APPENDIX-B
NECESSARY FIGURES





(a)



(b)

Figure B1: Geometry Factors for Helical Gears
(AGMA information sheet 225.01)

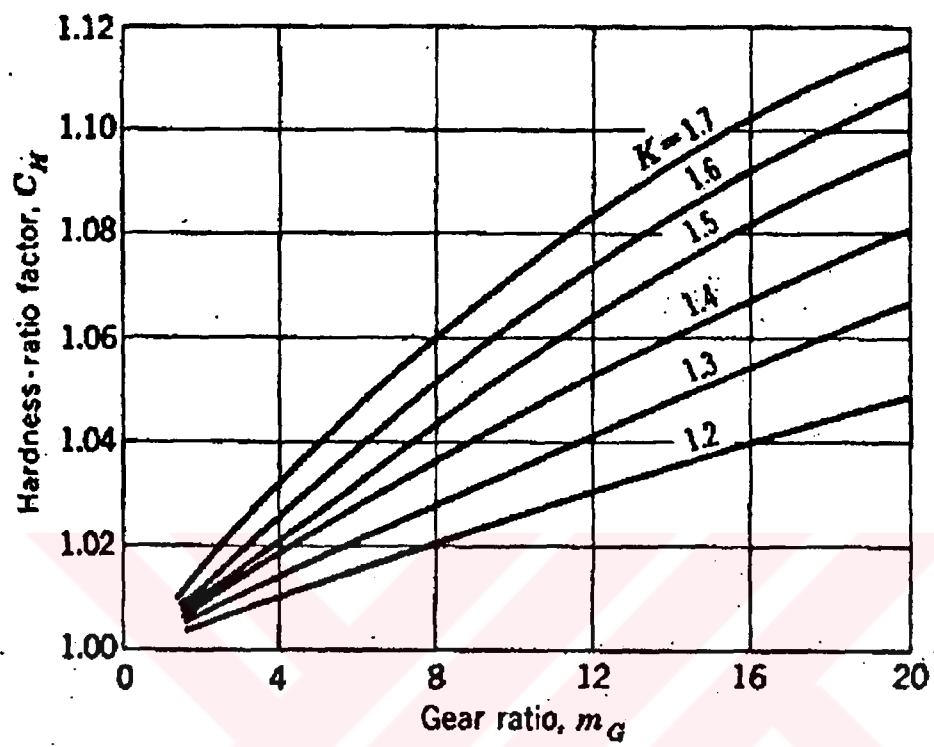


Figure B2: Hardness Ratio Factor (C_H) for Helical Gear