OPTIMISED PARAMETRIC GEAR SYSTEM DESIGN

by

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DECLARATION

The author declares that no part of this work has been submitted in support of another degree or qualification to this or any other establishment. The author further declares that he has not been a registered candidate or enrolled student for another award of this or other academic or professional institution during the course of the research program.

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I am also grateful that American Gear Manufacturers Association(AGMA) has allowed the public in general to quote and extract materials from AGMA standards. The official credit lines are acknowledged here and modified to cover all the references made: the AGMA formulae and diagrams used in this thesis are extracted from AGMA 218.01, AGMA 2001-B88 and AGMA 908-B89, with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.

Finally, I wish to thank my wife Mei and children Xu and RoJie for their support and understanding during the last four years while doing my PhD study.

ABSTRACT

This thesis is the summary of the research work that has been carried out by the author on the development of methods of optimum gear design and shaft design for gear boxes, and the integration of the methods into one software package together with parametric layout drawing.

The objective of the optimum gear design is to minimise the gear centre distance with a fixed aspect ratio of pinion, when power capacity requirement is given. Although power capacity is dependant upon many factors, the optimum gear design method has used module, numbers of teeth, and helix angle as the significant variables. The power capacity rating is calculated by AGMA standards and the optimum design method is based on a study of the numerical behaviour of the AGMA power capacity rating and transformation of the gear design constraints into direct limiting boundaries on design variables. Numerical example tests show that the method is efficient and effective in finding the global minimum centre distance design.

The shaft component design method is based on established theories for reaction calculation, bearing life rating, shaft stress calculation and shaft failure criterion. However, the layout of the gear box is defined by a unique system using vectors connecting shaft and gear centres. The gear design and shaft design methods are implemented in an integrated software package and a well defined data organisation provides the basis for data sharing.

The definition of layout by centre vectors also serves as the reference frame around which to draw the layout by parametric programs. The design results of the software package are obtained by parametric programs from a data file based on the same data organisation. Parametric programs for individual shaft, bearing, and gears are written to draw the components and main dimensions such as centre distances, component axial positions, gear sizes and bearing sizes are shown on the layout.

The contribution to knowledge by this investigation is mainly in the gear design area, viz., the study of the numerical behaviour of AGMA standards in relation to gear geometry and the development of the efficient and effective algorithm for the optimum gear design. The descriptive layout definition by centre vectors is also a novel feature.

NOMENCLATURE

Note: the symbols used in Chapter Two, Literature Survey, do not always comply with this nomenclature.

$_1$ = in subscript, for pinion

$_2$ = in subscript, for gear

a = shaft displacement vector

a^e = shaft element displacement vector

C = standard centre distance, inch(mm for numerical value)

C_1 = bearing dynamic capacity rating, N

C_{min}= minimum centre distance

C^L_{min}=lower boundary for the minimum centre distance

C^U_{min}=upper boundary for the minimum centre distance

C^L_{range}=lower limit of the centre distance range of (m_n,N_1,N_2)

C^U_{range}=upper limit of the centre distance range of (m_n,N_1,N_2)

C_r = operating centre distance, inch

C_a = application factor for pitting resistance

C_f = surface condition factor for pitting resistance

C_H = hardness ratio factor for pitting resistance

C_L = life factor for pitting resistance

C_m = load distribution factor for pitting resistance

C_p = elastic coefficeint, $psi^{1/2}$

C_R = reliability factor for pitting resistance

C_s = size factor for pitting resistance

C_T = temperature factor for pitting resistance

C_v = dynamic factor for pitting resistance

C_w = amalgamated factor for pitting resistance

d = operating pitch circle diameter of pinion, inch

= bearing bore size, mm

= shaft diameter, m

E = modulus of elasticity of shaft

E_1 = modulus of elasticity for pinion, psi

E_2 = modulus of elasticity for gear, psi

f = shaft force vector

f^e = shaft element force vector

F = face width, inch

F_a = bearing thrust load, N

F_e = equivalent dynamic load of bearing, N

F_r = bearing radial load, N

F_S = factor of safety for shaft design

h_a = addendum to module ratio

h_b = dedendum to module ratio

I = geometry factor for pitting resistance

I_z = moment of inertia of shaft cross section

J = geometry factor for bending strength

K = stiffness matrix of shaft segments

K^e = shaft element stiffness matrix

K_a = application factor for bending strength

K_B = rim thickness factor

K_f = stress correction factor

K_L = life factor for bending strength

K_m = load distribution factor for bending strength

K_R = reliability factor for bending strength

K_s = size factor for bending strength

K_T = temperature factor for bending strength

K_v = dynamic factor for bending strength

K_w = amalgamated factor for bending strength

L_{10}= bearing life in millions of revolutions

L_{10h}= bearing life in hours

L_e = length of a shaft segment(element)

L_{min}= minimum total length of face contact lines

m = transverse module, inch

m_{ar} = contact ratio derating factor

m_F = face contact ratio

m_G = gear ratio

m_N = load sharing ratio

m_n = normal module, inch(mm for numerical value)

$m^L{}_n$ = lower boundary for module in optimisation

$m^U{}_n$ = upper boundary for module in optimisation

m_p = transverse contact ratio

M = bending moment at a shaft section, Nm

n_a = fractional part of m_F

n_b = bearing speed, rpm

n_P = pinion speed, rpm

n_r = fractional part of m_p .

N = number of teeth

N_1 = number of teeth of pinion

N_2 = number of teeth of gear

N^L = lower boundary for number of teeth in optimisation

N^U = upper boundary for number of teeth in optimisation

P = transmitted power, hp

P_{ac}= allowable transmitted power for pitting resistance, hp

P_{at}= allowable transmitted power for bending strength, hp

P_d = transverse diametral pitch, inch^{-1}

P_{nd} = normal diametral pitch, nominal, inch^{-1}

p_b = transverse base pitch, inch

p_N = normal base pitch, inch

p_x = axial pitch, inch

Q_v = AGMA gear accuracy grade

r_b = fillet radius to module ratio

s_{ac} = allowable contact stress number, psi

s_c = contact stress number, psi

S_e = tensile endurance limit, N/m^2

s_{at} = allowable bending stress number, psi

s_t = bending stress number, psi

S_y = tensile yield strength, N/m^2

T = mean static torque at a shaft section, Nm

v_t = pitch line velocity, ft/min

W_a = gear axial load, lb

W_r = gear radial load, lb

W_t = transmitted tangential load, lb

x = addendum modification coefficient

Y = tooth form factor

Z = active length of line of action

δ = gear ratio tolerance

λ = pinion aspect ratio(face width/pitch circle diameter)

ϕ = standard transverse pressure angle

ϕ_n = standard normal pressure angle

ϕ_{nr} = operating normal pressure angle

ϕ_r = operating transverse pressure angle

μ_1 = Poisson's ratio for pinion

μ_2 = Poisson's ratio for gear

ψ = helix angle

ψ_b = base helix angle

ρ_1 = radius of curvature of pinion profile at pitch point

ρ_2 = radius of curvature of gear profile at pitch point

σ_a = alternating bending stress, N/m^2

τ_a = alternating torsional stress, N/m^2

τ_m = mean torsional stress, N/m^2

Σ = sum of

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CHAPTER ONE INTRODUCTION

1.1 *Gear Design and Design Automation*

Gears are widely used in industry to transmit power and change speeds between a driver and driven equipment. There are many types of gears, however external helical gears for use between parallel shafts are considered in this thesis. Helical gears are the most commonly used type of gears in heavy engineering for their high load carrying capacities and can be single helical or double helical, as shown in Fig.1-1. Fig.1-2 shows a typical reduction box of single helical gears for a steel rolling mill in which the input speed to the gear box from the motor is 500 rpm and the output speed from the gear box to the mill is 17.45 rpm. The output torque is 28.65 times the input torque since torque is inversely proportional to speed, when power loss in the gear box is considered insignificant. A motor generating the same output speed and torque would be much larger, less energy efficient and more costly than the combination of a small motor and a reduction gear box.

The design of power transmission gears involves selection of material, determination of geometry, deciding manufacturing method and precision, and consideration of mounting and lubrication to satisfy *the requirement of transmitting a specified power, at a given speed and for a required speed ratio*. The design of a gear train system will also include the design and selection of shafts and bearings.

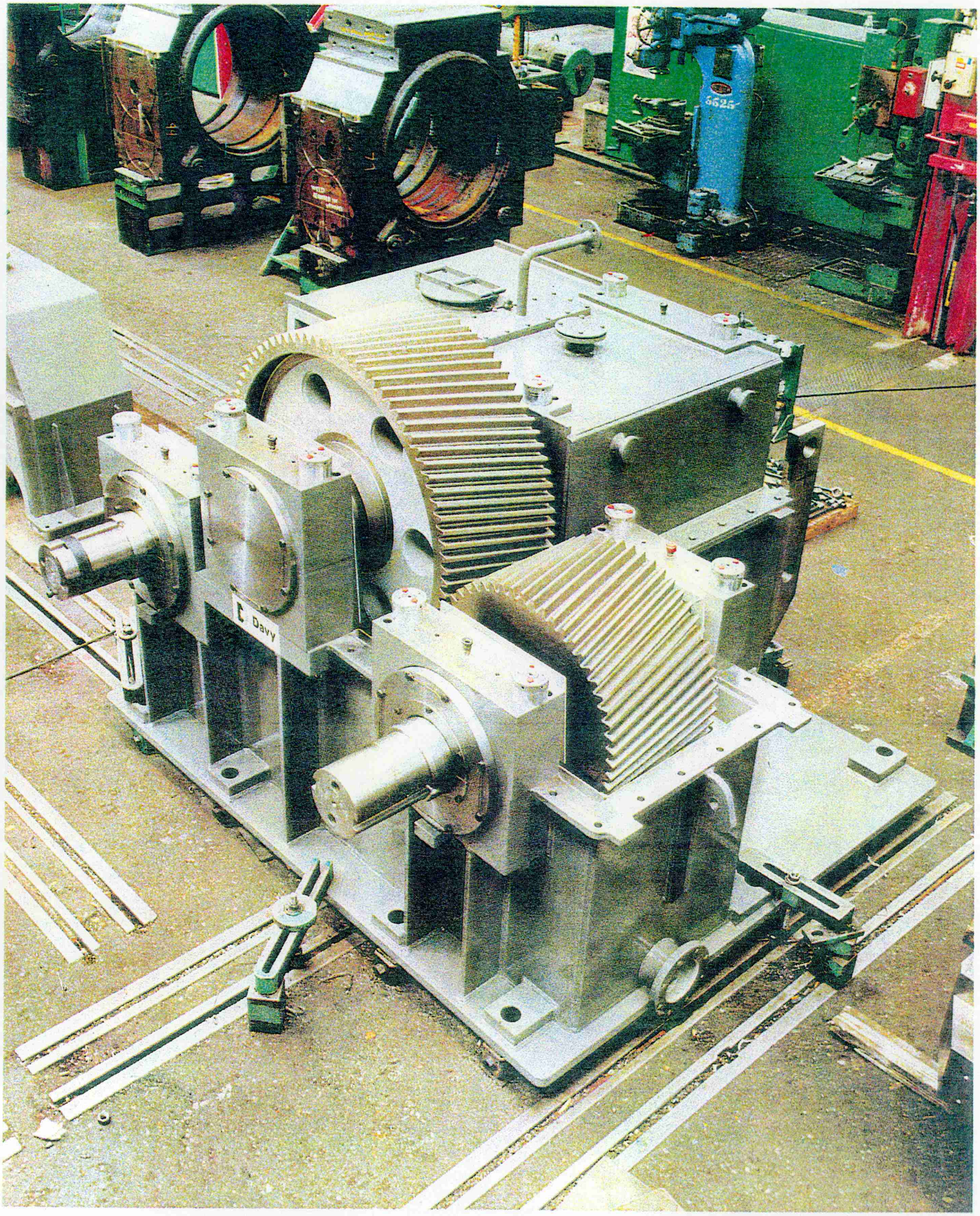


Fig.1-1a Single Helical Gears

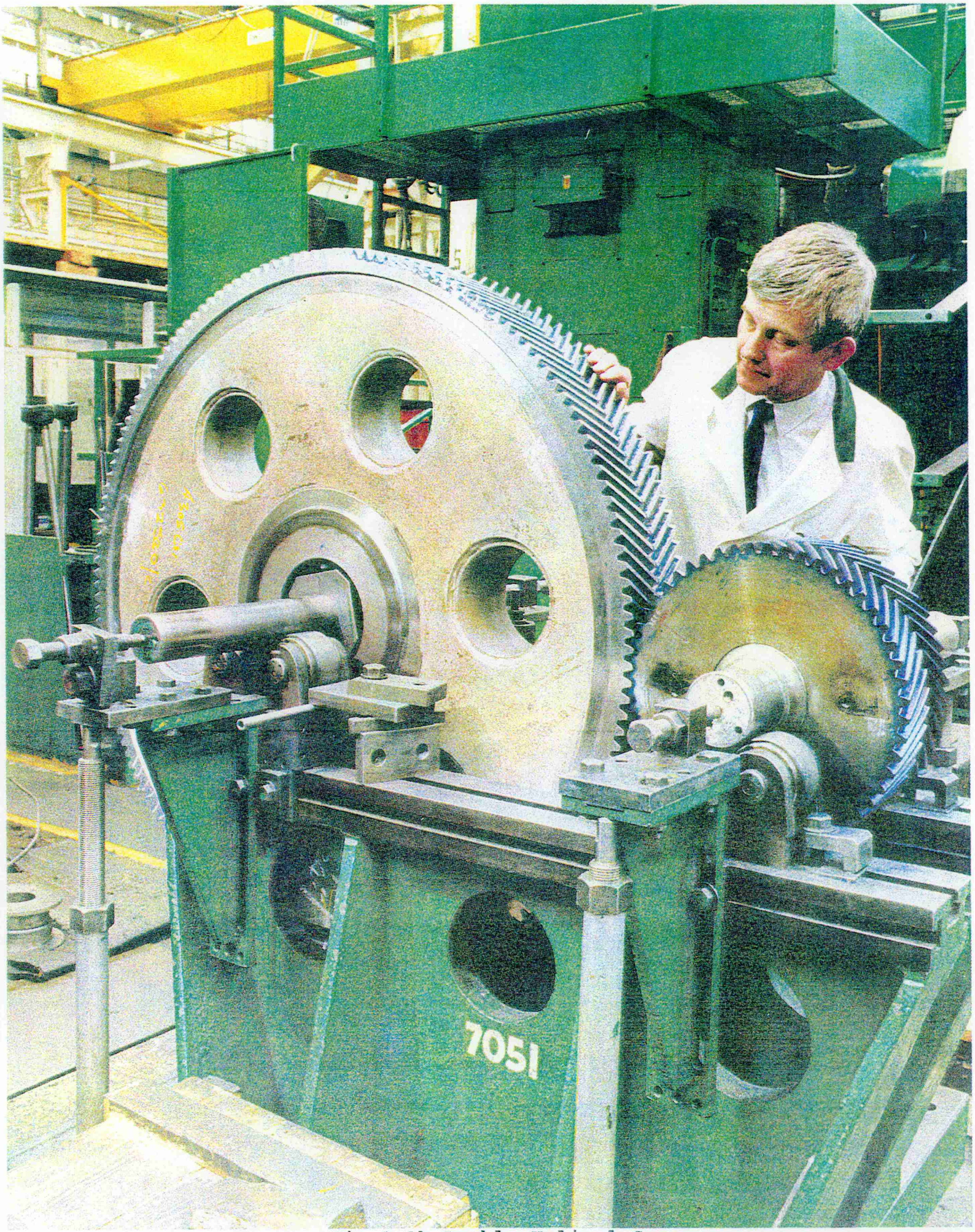


Fig.1-1b Double Helical Gears

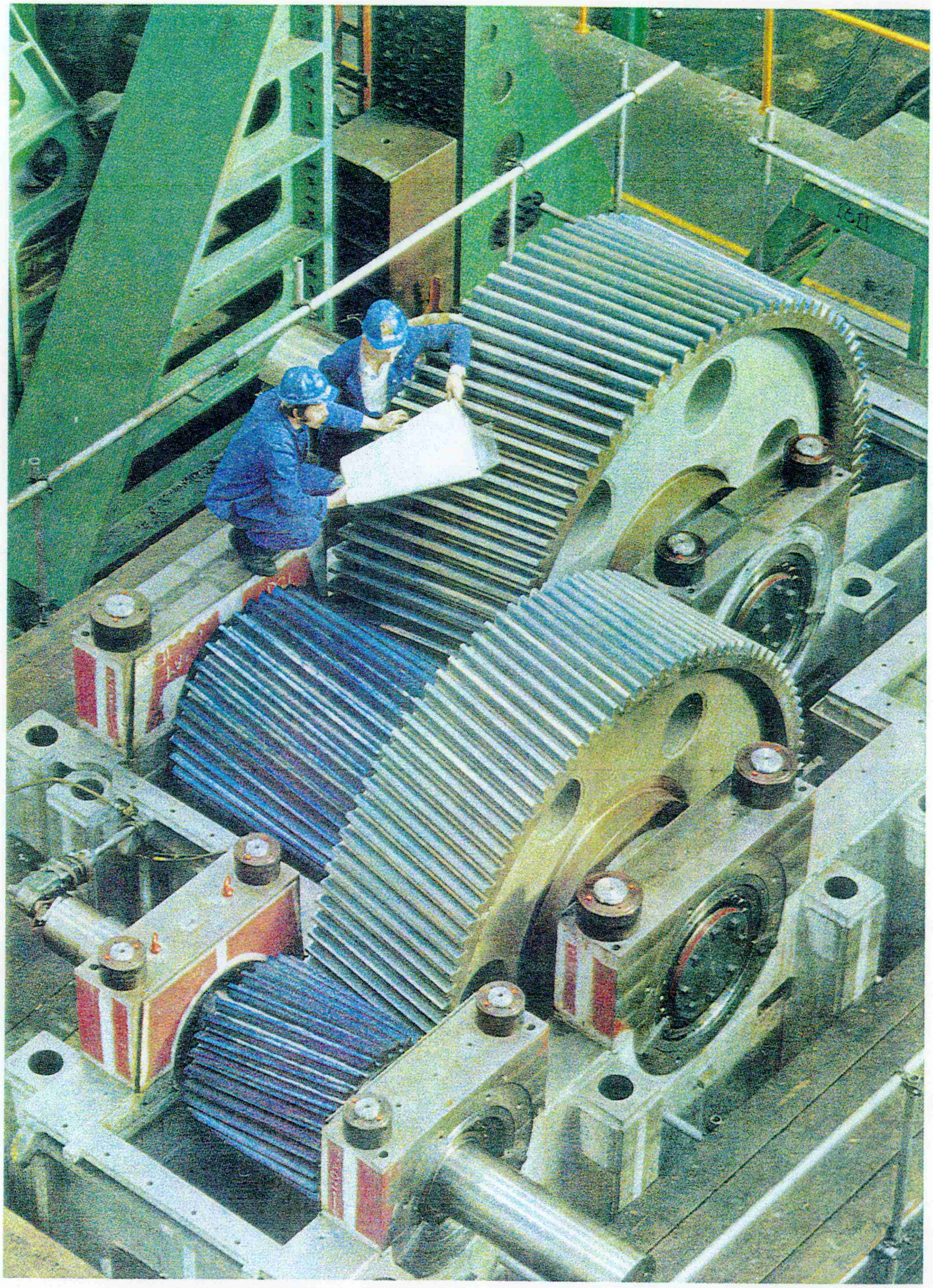


Fig.1-2a Photo of Double Reduction Gear Box

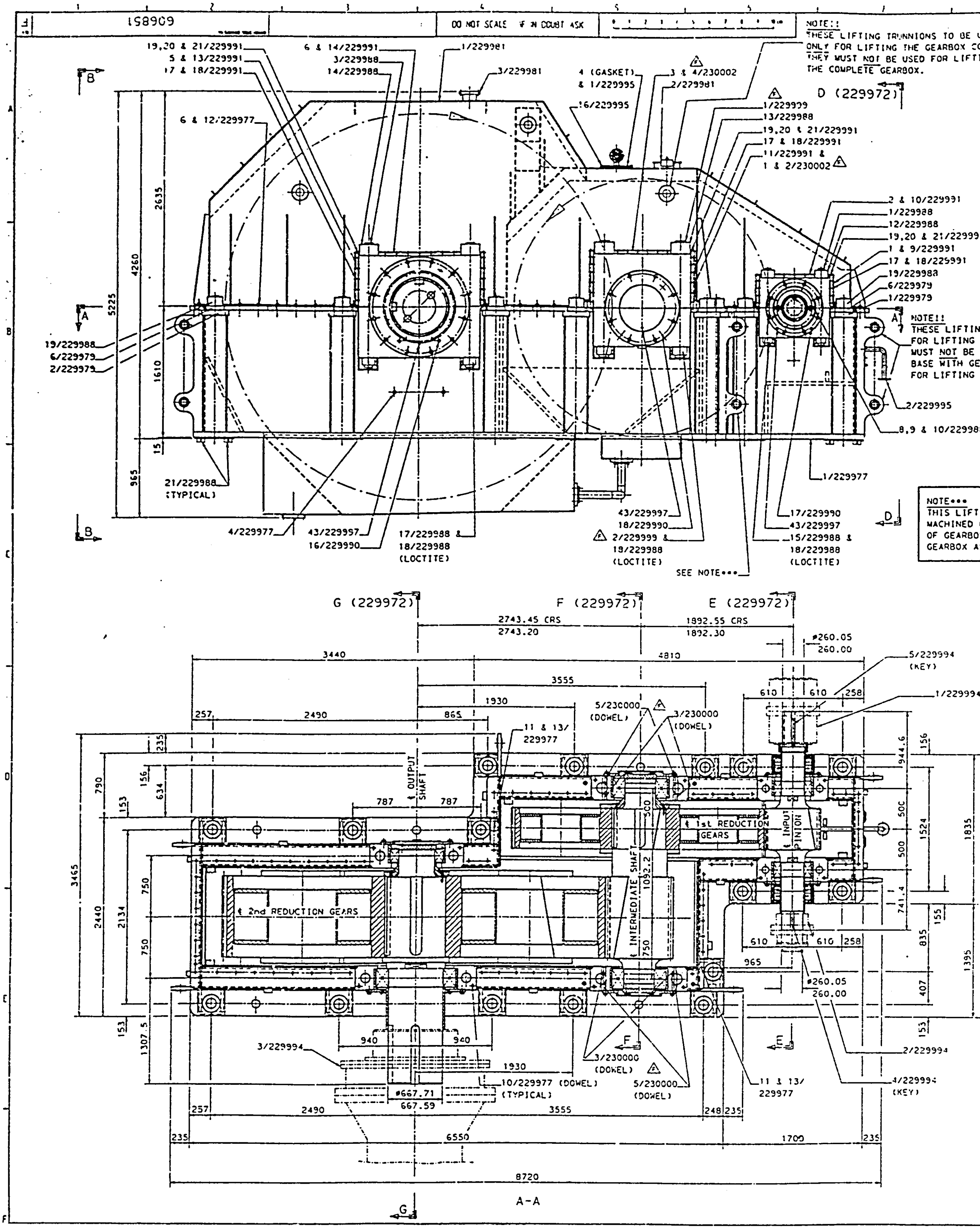


Fig.1-2b Drawing of Double Reduction Gear Box

Much is known about the power capacity rating of a set of gears when the material, gear geometry, manufacturing precision, mounting and lubrication conditions are known. The power capacity rating of a gear set is determined by the failure modes of the gear teeth. The two most common failure modes are gear tooth surface fatigue pitting and gear tooth root fatigue breakage. Gear tooth surface pitting is caused by Hertzian contact stress. Gear tooth root breakage is caused by bending stress in the tooth root. There are national and international standards for the calculation of gear capacity ratings, such as BS 436:Part3:1986 [17], AGMA 2001-B88 [5], and ISO 6336-1993 [33]. All of these standards specify lengthy and complex procedures for the calculation of allowable power capacity of a set of gears based on pitting resistance and bending strength. The standards are basically tools for analysing known gears, i.e., they are mathematical procedures or models that can be used to predict the power capacity of a set of known gears.

The design of a gear set is a reverse problem. The required power capacity and speed ratio are specified. Gear materials, geometry and other design parameters are to be determined to meet the power capacity and speed ratio requirements. The required power capacity acts as a design constraint in the sense that the allowable power capacity of the designed gear set must be larger than the specified value. The allowable power capacity of a candidate gear set has to be calculated by one of the standards to check whether the constraint of power requirement is satisfied. If the constraint is not satisfied, a new candidate will have to be determined and another round of checks is done. Different candidate gear sets will have to be tried until a satisfactory design is found. Obviously, there is a need for algorithms for gear design which will determine the candidate gear sets.

In gear design calculations, the determination of gear geometry is computationally intensive and of a repetitive and routine nature. It is very time consuming if done by longhand calculation. The traditional way of cut and try often results in less than optimum designs. The ever increasing pressure on the gear design engineer to design smaller and lighter gears in shorter time has caused a change in the way that gears are designed. Gear design by computer methods has become a necessity: not only can it help the engineer to design in a shorter time, but also achieve more economical designs by using optimisation methods.

Gear sets operate in the context of other rotating and supporting devices such as shafts and bearings. Gears have to be designed to work in harmony with them. Gear transmission shafts are designed for fatigue strength and also for stiffness. Large deflections have to be avoided because they cause load distribution over the gear teeth to deteriorate and reduce the power carrying capacity of the gears. Bearing size is determined by required bearing life and shaft diameters that it has to go through. Interferences between bearing outside diameters of two closely placed gear shafts must be avoided. For a multiple reduction gear box, interferences between gear and shaft outside diameters must

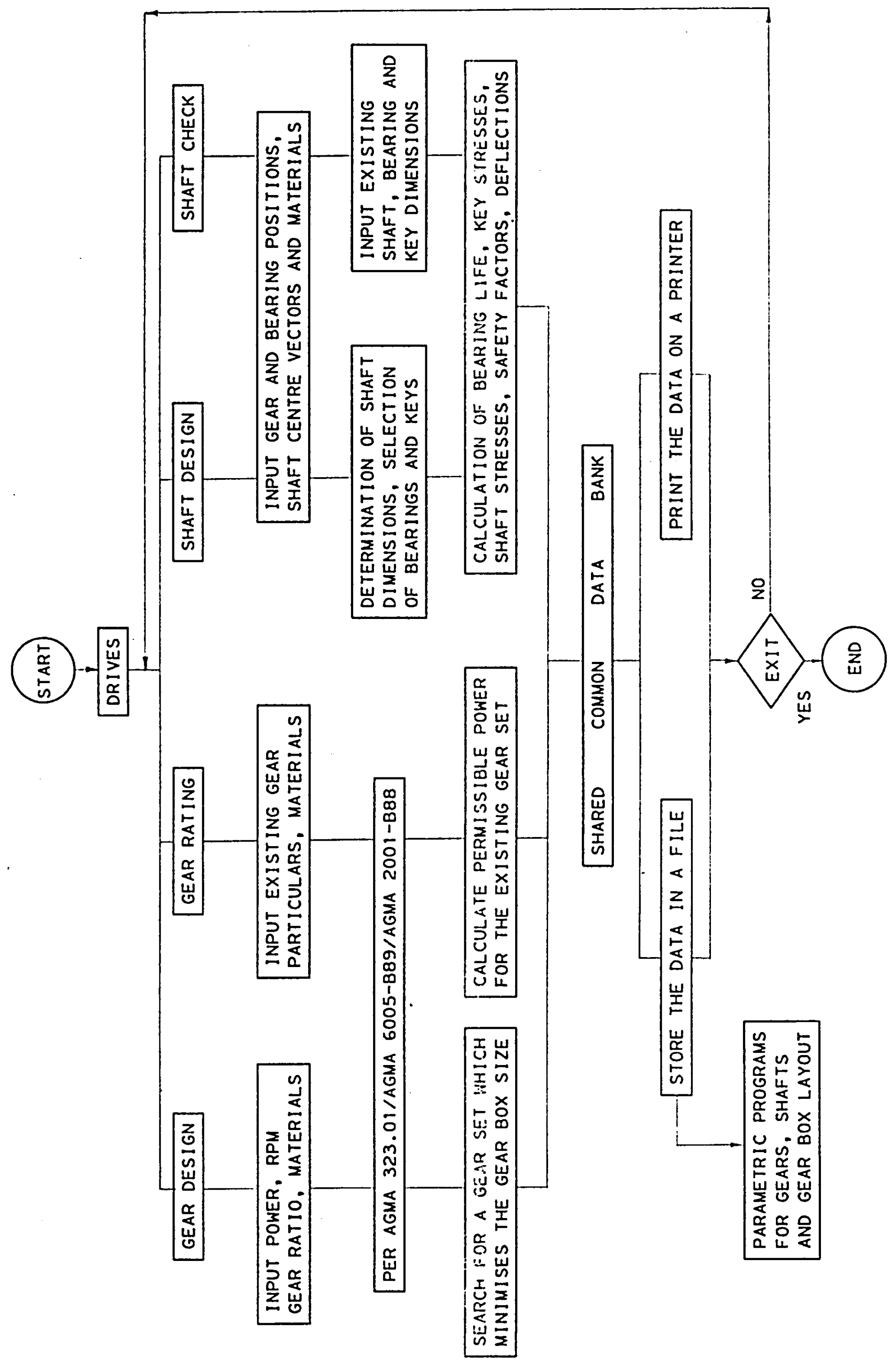


Fig.1-3 Functional Block Diagram

also be avoided. The computerising of shaft design and bearing selection in the same software package as for gear design will make it much easier to achieve a consistent and balanced design for a gear box.

Drawing is the most concise and informative language in engineering. A design in mere numerical figures is not as well presented as in a drawing. It is even more so when it is a layout drawing of a gear box showing not only dimensions but also geometric relations. To draw such a drawing automatically on computer helps the engineer to see the overall design immediately and decide if any modification is necessary.

To summarise, the aim of the investigation is:

1) to optimise gear sets by computer methods and to computerise the design of shafts and bearings;
2) to generate the layout drawing of the optimised gear train on computer automatically by parametric programs.

The project is sponsored by Davy International, Sheffield. A software package for gear system design is developed on the company's CAE system. The package is based on the outcome of the investigation and the company's experience in heavy engineering gear design and oriented towards this type of gear design(see Fig.1-3, Functional Block Diagram).

1.2 *Basic gear geometry and terminology*

The gear industry, like many other industries, uses a lot of terms which concisely express the special meanings contained in them. ANSI/AGMA 1012-F90[8], Gear Nomenclature, Definition

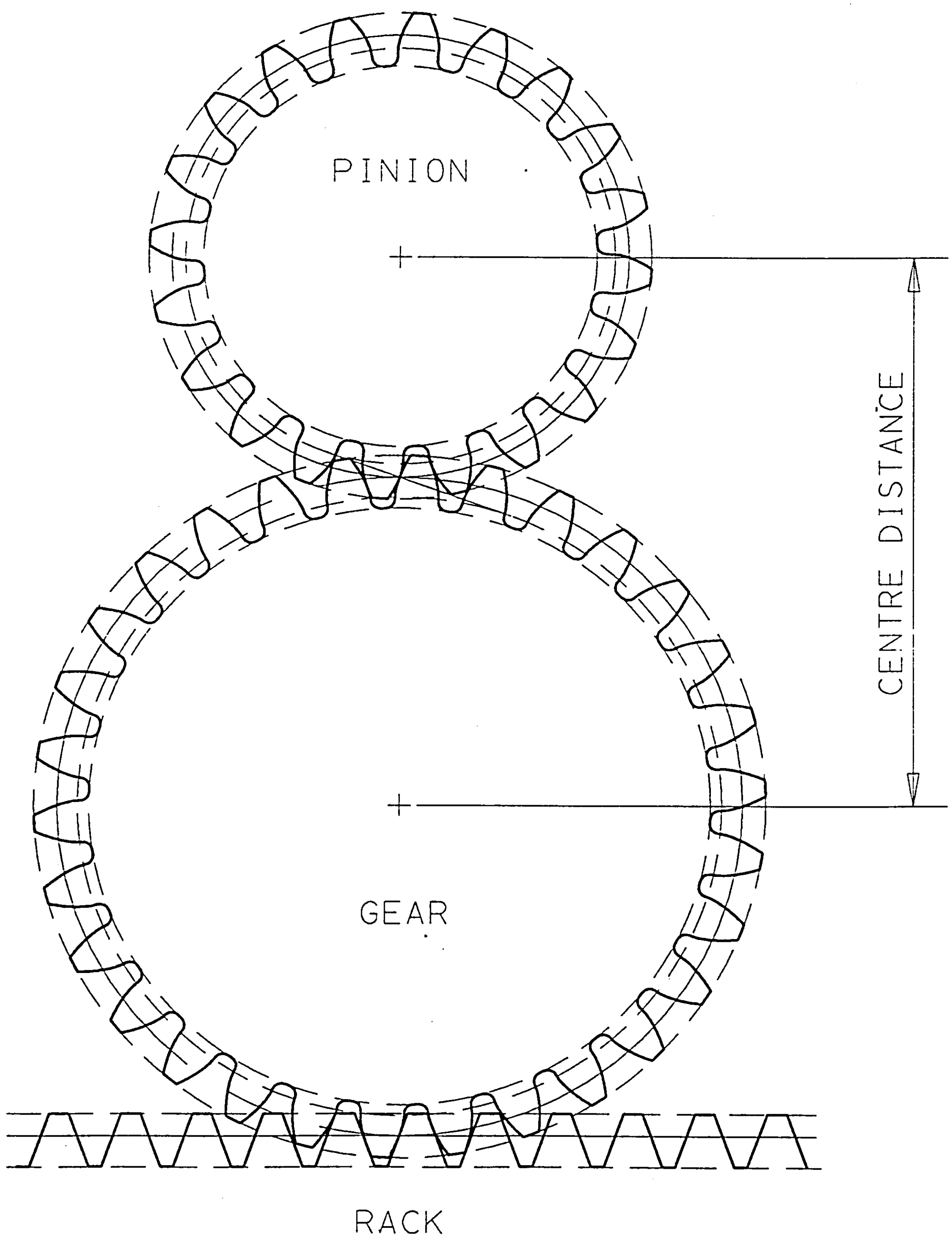


Fig.1-4 Gears in Mesh

of Terms with Symbols, provides the agreed definitions of terms, symbols and abbreviations used by the gear industry. In this section, only the basic gear geometry and terminology used in this thesis are explained, using ANSI/AGMA 1012-F90 as a guide.

Gears transmit motion by means of successively engaging teeth, as shown in Fig.1-4. For power transmission gears, *involute* profile teeth are most commonly used. An involute curve can be created by unwrapping a cord around a circle as shown in Fig.1-5. The circle is called *base circle* and any fixed point on the chord, e.g. point b, will trace out an involute curve. The instantaneous radius of curvature at point b is eb. Line eb is equal to arc ae, normal to the involute at point b and tangent to the base circle at point e.

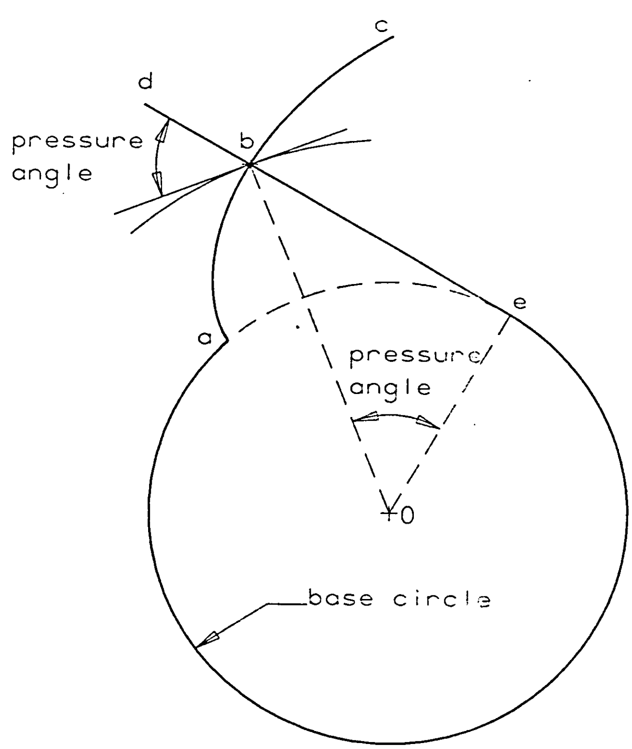


Fig.1-5 Generation of an Involute

When the teeth of two gears are engaged, as shown in Fig.1-6, a common tangent to the two base circles can be drawn. The common tangent is also the common normal of tooth profile at points of contact. This common tangent is called the *line of action* because all the contact points move along the line as the two gears rotate. The two gears come into mesh at point A and come out of mesh at point B. Points A and B are intersections of the O/D's of the two gears with the line of action. The length of line segment between A and B is called the *active length of line of action* Z. *Pitch point* is the intersection of the line of action and the line connecting the centres of the two gears. Two circles tangent at the pitch point and centred at the gear centres are called *pitch circles*. The meshing action starting from point A till the pitch point is called *approach action* and that from pitch point till the end point B is called *recess action*. Since there is no involute curve below the base circle, the tooth tip of one gear meshing with the tooth root of the other gear will have *involute interference* if the meshing point is below the base circle. Involute interference would happen if the geometries of the two gears are such that point A or point B lies beyond the tangent points of line of action with the base circles.

As a contact point moves along the line of action, the speed vectors of the two gears at the contact point is generally different except when the contact point is at the pitch point. This means that the relative motion between the profiles of the teeth of the two gears is generally a

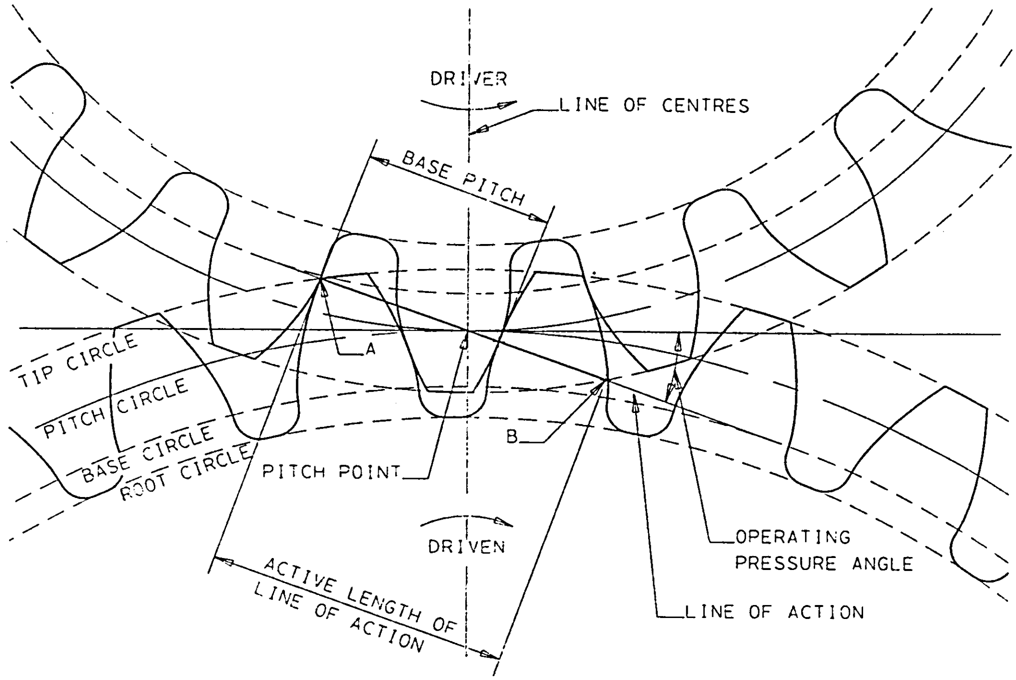


Fig.1-6 Teeth in Engagement

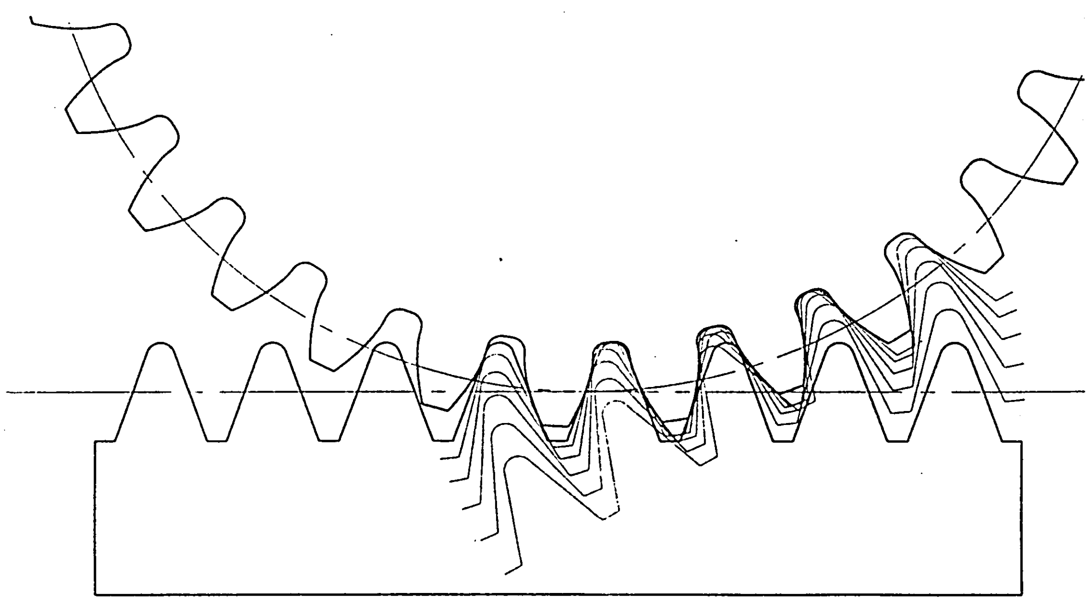


Fig.1-7 Generation of Gear Teeth by Rack Type Cutter

combination of rolling and sliding, with pure rolling only at pitch point. *Sliding ratio* is the ratio of sliding velocity to rolling velocity. It is a measure of the severity of the sliding motion and is generally different for the two gears at the same contact point.

Operating centre distance C_r is the distance between the centres of the two gears. *Face width* F is the axial length of the gear teeth. Of the two gears, *pinion* is the smaller one and the larger one is called *gear*. *Gear ratio* m_G is the ratio of the number of gear teeth N_2 to the number of pinion teeth N_1. A *rack* can be imagined as a gear with an infinitely large pitch circle. The pitch circle of a rack becomes a pitch line and the involute profile becomes a straight line profile.

Gears can be *generated* by meshing with a rack type cutter, as shown in Fig.1-7. When the geometry of the gear being generated is such that involute interference with the cutter would happen, the tip of the gear cutting tool will cut out a recess which is called *undercut* near the root of tooth flank. Undercut can happen when the number of teeth of the gear being cut is small. When a gear has been generated by a rack type cutter, interference with another gear will not happen because the material has already been removed by undercut. Undercut will make the tooth root weaker. By adopting a minimum number of teeth, undercut or interference can be avoided.

The terminology of helical gear teeth is illustrated in Fig.1-8. The *standard pitch circle* is a theoretical circle upon which all the nominal values are based. The sum of the

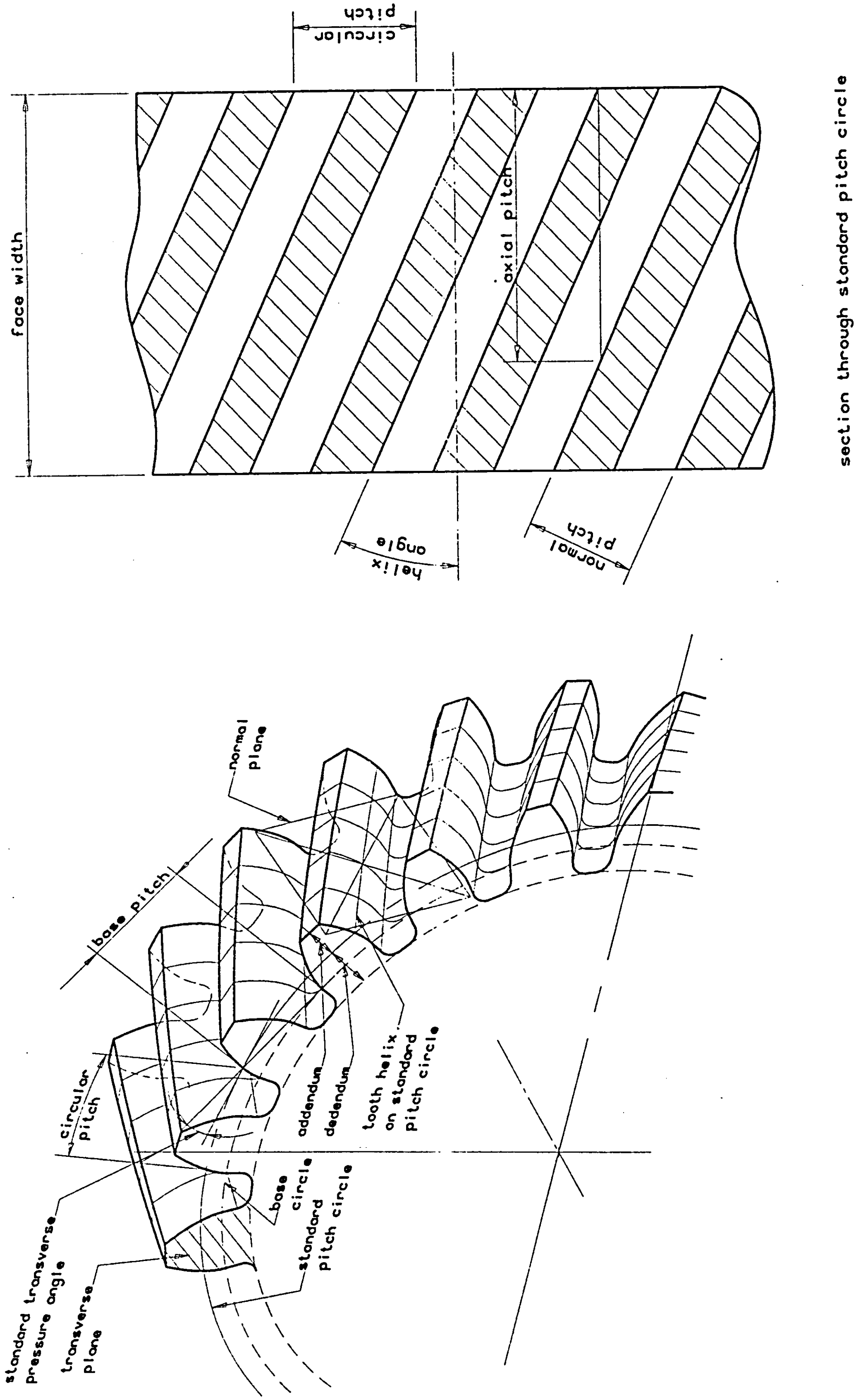


Fig.1-8 Helical Gear Terminology

radii of standard pitch circles of pinion and gear is called *standard centre distance* C. The tooth profile of helical gears is an involute curve in the *transverse* plane which is the plane of rotation. The *normal* plane is normal to the tooth helix on the standard pitch circle.

Pressure angle is the angle between the normal of tooth profile at a circle concerned and the tangent of the circle at the same point on the profile, as shown in Fig.1-5. The pressure angle that is defined on the standard pitch circle is called the *standard normal pressure angle* ϕ_n in the normal plane and *standard transverse pressure angle* ϕ in the transverse plane. The most commonly used standard normal pressure angle ϕ_n is 20 degrees. Standard normal pressure angles of 14.5, 22.5 and 25 degrees are occasionally used. Operating pressure angle is defined on the pitch circle, as shown in Fig.1-6. *Operating normal pressure angle* is defined in the normal plane and designated ϕ_{nr}. *Operating transverse pressure angle* is defined in the transverse plane and designated ϕ_r. *Helix angle* ψ is the angle between the tooth helix and a line parallel to the gear axis on the standard pitch circle. *Base helix angle* ψ_b is the angle of the tooth helix on the base circle.

The *transverse module* m is the ratio of the standard pitch circle diameter to the number of teeth. *Normal module* m_n is the value of module in the normal plane. The module is the index of tooth size in metric units. The reciprocal of the module, call the *diametral pitch* P_d, is used as a measure of tooth size in inch units. Values of module and diametral pitch have been standardised to save on the costs of tooling and manufacturing.

Pitch is the distance between the same flank of two adjacent teeth along a certain direction. The pitch defined on the standard pitch circle is called *circular pitch*. The pitch defined on base circle or along the line of action is called *base pitch*. The base pitch defined in transverse plane is called *transverse base pitch* p_b and that defined in normal plane is called *normal base pitch* p_N. The pitch along the axis of the gear is called *axial pitch* p_x. *Transverse contact ratio* m_p is the ratio of the active length of line of action to the transverse base pitch. *Face contact ratio* m_F is the ratio of the face width to the axial pitch. Helical gears with a face contact ratio $m_F > 1$ is called *conventional helical gears* and is the predominant gear type used in heavy engineering.

Addendum is the radial distance between the tip circle and the standard pitch circle. *Dedendum* is the radial distance between the root circle and the standard pitch circle. *Gear tooth system* refers to the group of parameters of standard pressure angle, addendum to module ratio, dedendum to module ratio, and fillet radius to module ratio.

When gears are cut by a method of generation, e.g. by a rack type cutter, the gear cutter can be moved out or in from the nominal position relative to the gear centre, as shown in Fig.1-9. This movement of the cutter is called *addendum modification* and will cause a shift in the involute profile of the teeth. Positive addendum modification moves the cutter out and negative in. The ratio of addendum modification to module is called *addendum modification coefficient* x. When the sum of addendum modification coefficients of pinion and gear $\Sigma x=0$, operating centre distance and pressure angle are the same as standard centre distance and pressure angle, i.e., $C_r=C$ and $\phi_r=\phi$.

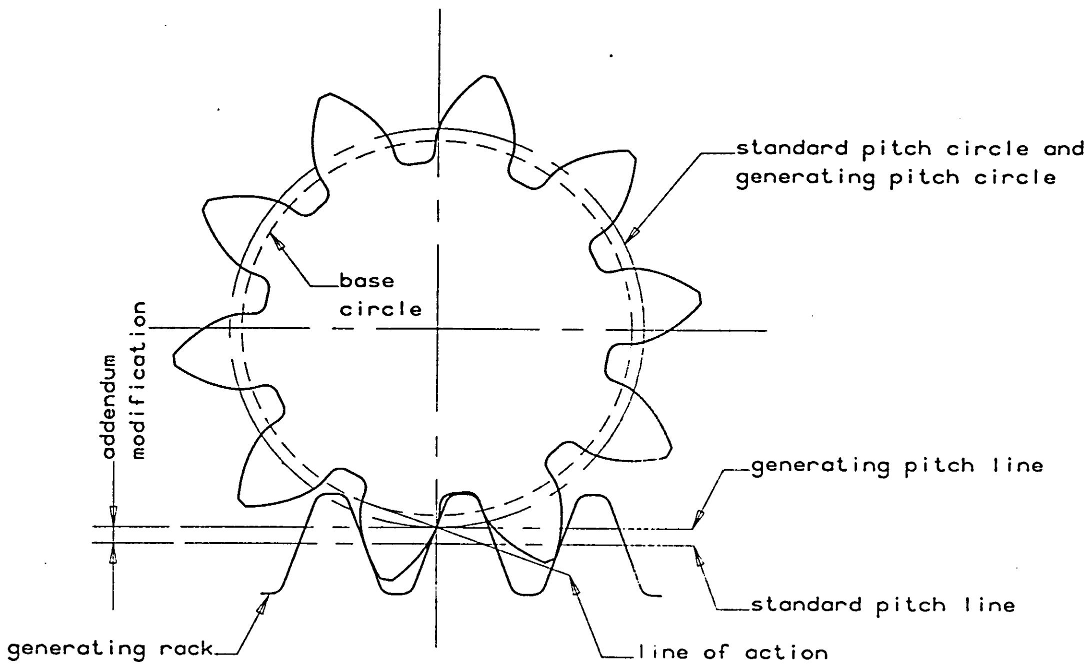


Fig.1-9 Addendum modification

1.3 *Basic Relations of Parameters and Variables*

The determination of the geometry of helical gears involves determination of such variables as centre distance, face width, module, numbers of teeth of pinion and gear, helix angle, and addendum modification coefficients. Standard pressure angle, addendum to module ratio, dedendum to module ratio, and fillet radius to module ratio are treated as fixed design parameters, since there are only a small number of standardised gear tooth systems. The following gives the more useful relations of gear parameters and variables.

Normal module:

$$m_n = m\cos\psi \tag{1-3-1}$$

Normal diametral pitch:

$$P_{nd} = \frac{1}{m_n} \tag{1-3-2}$$

Transverse diametral pitch:

$$P_d = P_{nd}\cos\psi \tag{1-3-3}$$

Pitch circle diameter of pinion:

$$d = \frac{2C_r}{m_G + 1} \tag{1-3-4}$$

Standard centre distance:

$$C = \frac{(N_1 + N_2)m_n}{2\cos\psi} \tag{1-3-5}$$

Transverse contact ratio:

$$m_p = \frac{Z}{p_b} \tag{1-3-6}$$

Face contact ratio:

$$m_F = \frac{F}{p_x} \tag{1-3-7}$$

Axial pitch:

$$p_x = \frac{\pi m_n}{\sin\psi} \tag{1-3-8}$$

Pitch line velocity:

$$v_t = \frac{\pi d n_p}{12} \tag{1-3-9}$$

Transverse pressure angle:

$$\tan\phi = \frac{\tan\phi_n}{\cos\psi} \tag{1-3-10}$$

Base helix angle:

$$\sin\psi_b = \sin\psi\cos\phi_n \tag{1-3-11}$$

1.4 *Gear Capacity Rating by AGMA Method*

An exercise of gear capacity rating is aimed at ensuring that the gears concerned do not fail in service. There are many ways that a set of gears can fail. AGMA 110.04 [3], Nomenclature of Gear Tooth Failure Modes, identifies and describes classes of common gear failures. Nevertheless, gear capacity rating by any of the national or international gear rating standards is based only on the two most common gear failure modes, i.e., gear tooth surface pitting and gear tooth root fatigue breakage. Scoring is another surface failure caused by high contact stress, temperature and sliding on the tooth flank, near tooth tip or root. Although scoring is checked for some type of gear applications, no consensus of opinions has been reached for a method to be given in the gear rating standards. The rating methods given below comes from AGMA 2001-B88, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.

1.4.1 *Basic Formulae*

1.4.1.1 *Pitting Resistance*

The AGMA pitting resistance formula is based on the Hertz contact stress equation for cylinders with parallel axes. Two teeth of a pinion and a gear are shown in contact in Fig.1-10(on page 22). The radii of the cylinders are the radii of curvature of the teeth at the point of contact. The fundamental pitting resistance formula derived by AGMA is

$$s_c = C_p \left(\frac{W_t}{d\,F\,I} C_a \frac{C_m}{C_v} C_s C_f \right)^{1/2} \qquad (1\text{-}4\text{-}1)$$

where

s_c = contact stress number, psi

C_p = elastic coefficient, $psi^{1/2}$

W_t = transmitted tangential load, lb

d = operating pitch circle diameter of pinion, in

F = face width, in

I = geometry factor for pitting resistance

C_a = application factor for pitting resistance

C_m = load distribution factor for pitting resistance

C_v = dynamic factor for pitting resistance

C_s = size factor for pitting resistance

C_f = surface condition factor for pitting resistance

The elastic coefficient in equation (1-4-1) is

$$C_p = \left\{ \frac{1}{\pi\left[\left(\frac{1-\mu_1^2}{E_1}\right)+\left(\frac{1-\mu_2^2}{E_2}\right)\right]} \right\}^{1/2} \tag{1-4-2}$$

where

μ_1 = Poisson's ratio for pinion

μ_2 = Poisson's ratio for gear

E_1 = modulus of elasticity for pinion, psi

E_2 = modulus of elasticity for gear, psi

The transmitted tangential load in equation (1-4-1) is

$$W_t = \frac{126000\ P}{n_p\ d} \tag{1-4-3}$$

where

P = transmitted power, hp

n_p = pinion speed, rpm

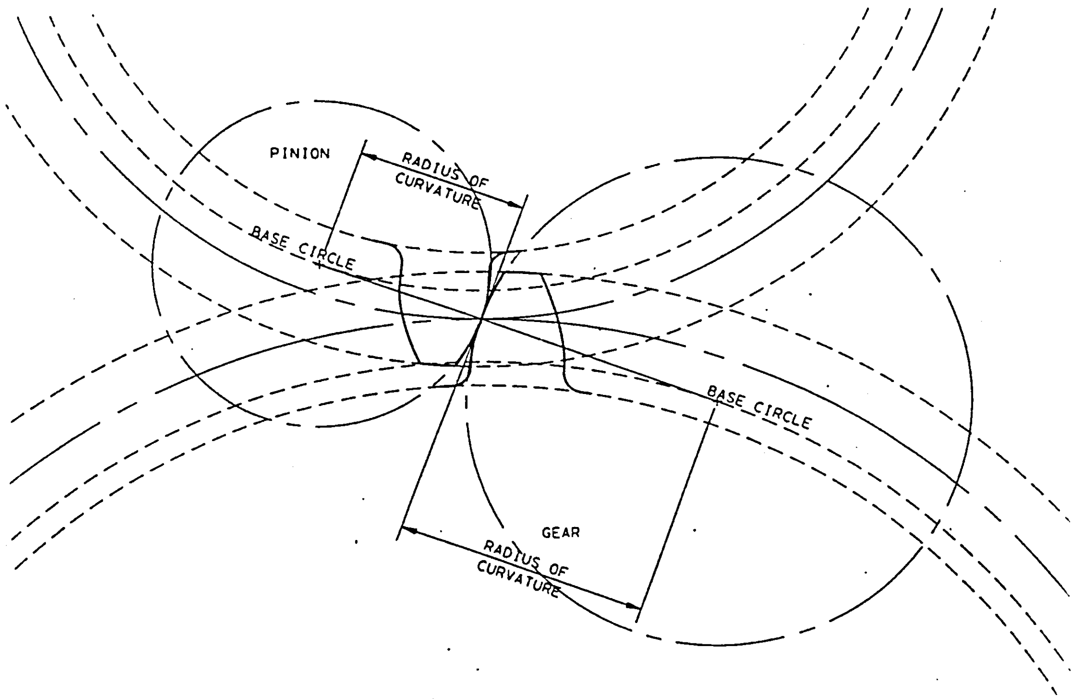


Fig.1-10 Teeth in contact

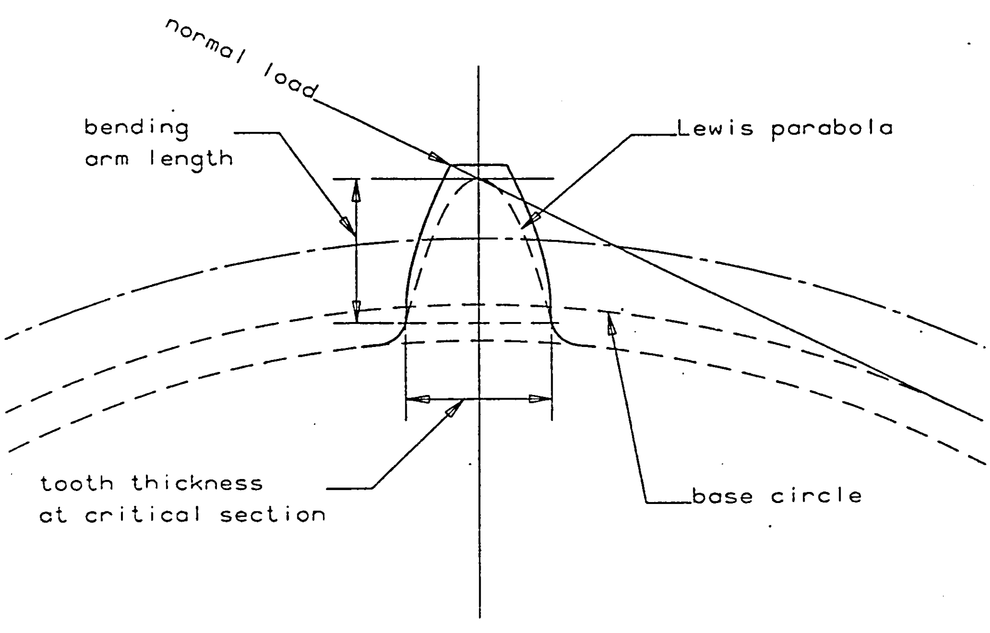


Fig.1-11 Tooth under bending

The calculated contact stress number s_c must be less than or equal to the modified allowable contact stress number s_{ac} as follows:

$$s_c \leq s_{ac} \frac{C_L \; C_H}{C_T \; C_R} \qquad (1\text{-}4\text{-}4)$$

where

s_{ac} = allowable contact stress number, psi

C_L = life factor for pitting resistance

C_H = hardness ratio factor for pitting resistance

C_T = temperature factor for pitting resistance

C_R = reliability factor for pitting resistance

The AGMA power rating formula for pitting resistance is a combination of equations (1-4-1), (1-4-3) and (1-4-4), as follows,

$$P_{ac} = \frac{n_P \; F}{126000} \; \frac{I \; C_v}{C_s \; C_m \; C_f \; C_a} \left(\frac{d \; s_{ac}}{C_p} \; \frac{C_L \; C_H}{C_T \; C_R} \right)^2 \qquad (1\text{-}4\text{-}5)$$

where P_{ac} is the allowable transmitted power for pitting resistance, hp.

1.4.1.2 *Bending Fatigue Strength*

The AGMA formula for bending fatigue strength at tooth root is based on the method of Lewis[35] where the gear tooth is simplified as a cantilever beam of uniform strength with the shape of a parabola inscribed within the tooth, as shown in Fig.1-11. The root stress is then corrected for stress concentration using the results of photoelastic experiments of Dolan and Broghamer[27]. The fundamental formula for bending stress in gear tooth root derived by AGMA is,

$$s_t = \frac{W_t \; P_d}{F \; J} K_a \frac{K_m}{K_v} K_s K_B \tag{1-4-6}$$

where

s_t = bending stress number, psi

P_d = transverse diametral pitch, in^{-1}

J = geometry factor for bending strength

K_a = application factor for bending strength

K_m = load distribution factor for bending strength

K_v = dynamic factor for bending strength

K_s = size factor for bending strength

K_B = rim thickness factor

The calculated bending stress number s_t must be less than or equal to the modified allowable bending stress number s_{at} as follows:

$$s_t \le \frac{S_{at} \; K_L}{K_T \; K_R} \tag{1-4-7}$$

where

s_{at}= allowable bending stress number, psi

K_L = life factor for bending strength

K_T = temperature factor for bending strength

K_R = reliability factor for bending strength

The AGMA power rating formula for bending strength is a combination of equations (1-4-3), (1-4-6) and (1-4-7), as follows,

$$P_{at} = \frac{n_p \; F}{126000} \frac{d}{P_d} \frac{J \; K_v}{K_s \; K_m \; K_B \; K_a} \frac{s_{at} \; K_L}{K_R \; K_T} \tag{1-4-8}$$

where P_{at} is the allowable transmitted power for bending strength, hp.

The power capacity rating formulae for pitting resistance (1-4-5) and for bending strength (1-4-8) must be applied to pinion and gear separately. The power capacity of a gear set is the lowest of the four capacities calculated by equations (1-4-5) and (1-4-8) for pinion and gear.

1.4.2 *Calculation Factors*

The calculation factors in the AGMA rating formulae can be divided into two groups. The first group includes the geometry factors I and J, and the elastic coefficient C_p. These factors are essential factors in the sense that they simply represent more complicated formulae or calculating procedures. The introduction of these factors can make the presentation of the rating formulae simpler and more meaningful. The second group includes all the other factors. The factors starting with the letter C are pitting resistance calculation factors and those starting with the letter K are bending strength calculation factors. The factors in the second group are correcting factors in the sense that they modify the nominal situation to the actual situation.

The geometry factors I and J, as the names imply, are functions of tooth geometry. The determination of geometry factors I and J are complex and comes from AGMA Information Sheet 908-B89 [7], a numerical procedure recently developed as an improvement on the graphical approach in the previous AGMA 218.01 [4].

The pitting resistance geometry factor I takes into account the relative combined radius of curvature of pinion and gear at the contact stress calculation point and load sharing between pairs of teeth simultaneously in mesh. The formula for the calculation of geometry factor I given in AGMA 908-B89 can be written in a simpler form for conventional helical gears as follows,

$$I = \frac{\cos\phi_r}{\left(\frac{1}{\rho_1} + \frac{1}{\rho_2}\right) d\, m_N} \qquad (1\text{-}4\text{-}9)$$

where

ϕ_r = operating transverse pressure angle

ρ_1 = radius of curvature of pinion profile at pitch point

ρ_2 = radius of curvature of gear profile at pitch point

m_N = load sharing ratio

A numerically equivalent formula given in AGMA 218.01 can also be written in a simpler form for conventional helical gears as follows,

$$I = \frac{\cos\phi_r \sin\phi_r}{2\, m_N} \; \frac{m_G}{m_G + 1} \qquad (1\text{-}4\text{-}10)$$

The bending strength geometry factor J takes into account more variables. They are the tooth root thickness, the length of bending arm of the inscribed Lewis parabola, the effect of oblique helical line loading, fillet stress concentration and load sharing between pairs of teeth. The formula for the geometry factor J as given in AGMA 908-B89 can also be simplified for the conventional helical gears and it is

$$J = \frac{Y}{K_f \, m_N} \qquad (1\text{-}4\text{-}11)$$

where

Y = tooth form factor

K_f= stress correction factor

More detailed discussion on the geometry factors I and J is presented in Chapter Four.

The value of the elastic coefficient C_p for steel gears is 2300 psi$^{1/2}$. Values for other combinations of gear materials can be found by using equation (1-4-2).

In the second group, application factors C_a and K_a, load distribution factors C_m and K_m, and dynamic factors C_v and K_v, are the more variable ones. Life factors C_L and K_L, and reliability factors C_R and K_R may need to be considered if limited life and reliability other than 99% is required. Size factor C_s and K_s, temperature factors C_T and K_T, and surface condition factors C_f can be assumed to be 1. Hardness ratio factor C_H and rim thickness factor K_m often take values of 1, but if different materials are used or gear blank rim is thin, other values may be used for C_H and K_m.

The application factors C_a and K_a make allowance for any externally applied loads in excess of the nominal tangential load W_t. The application factors are established after considerable field experience has been gained for a particular application. Values for application factors should be determined before the design of gears. In the absence of more specific data, values for application factors given in AGMA 6010-E88 [6], Standard for Spur, Helical, Herringbone, and Bevel Enclosed Drives, can be used. The values given in AGMA 6010-E88 range from 1.25 to 3.50 and cover a wide range of applications from cereal cooker to metal processing mill.

The dynamic factors C_v and K_v account for internally generated gear tooth loads which are induced by non-conjugate meshing action of the gear teeth. C_v and K_v are defined as the ratio of transmitted tangential load to the total of transmitted tangential load and internally generated gear tooth load. The values of dynamic factors are related to AGMA gear accuracy grade Q_v and pitch line velocity, as shown in Fig.1-12. For a first approximation, finish hobbed gears can normally achieve Q_v=8 and ground gears Q_v=11. Closed form formulae are also given by AGMA, as follows.

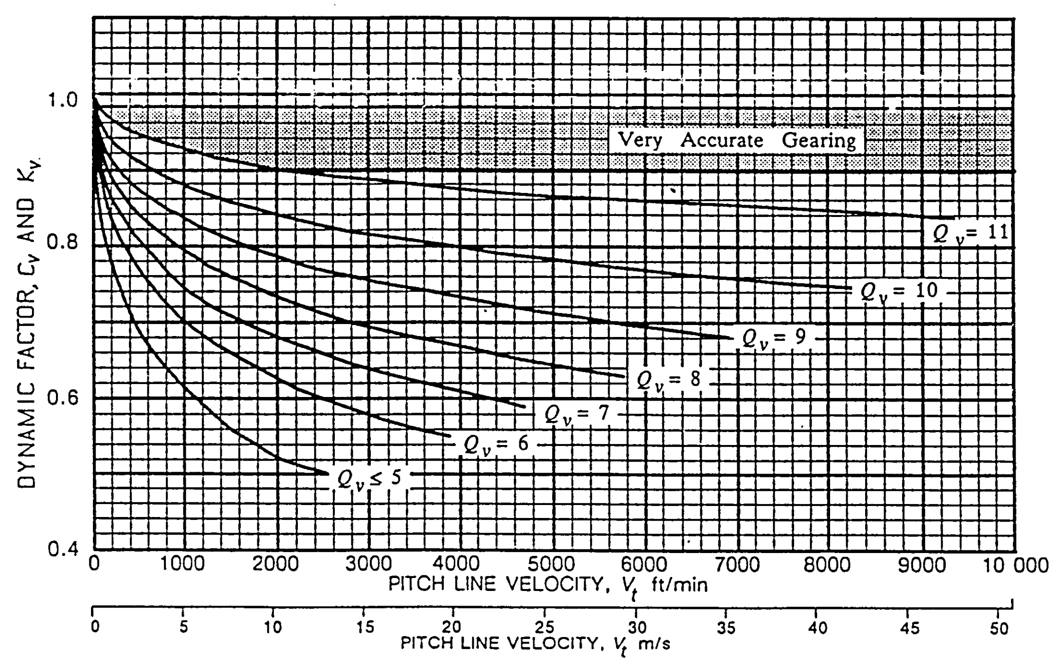


Fig.1-12 Dynamic Factors, C_v and K_v for different Q_v(AGMA gear accuracy grades)

$$C_v = K_v = \frac{50}{50 + v_t^{1/2}} \tag{1-4-12}$$

for AGMA gear accuracy grade 5, and

$$C_v = K_v = \left(\frac{A}{A + v_t^{1/2}}\right)^B, \qquad v_t \leq (v_t)_{max} \tag{1-4-13}$$

for AGMA gear accuracy grade 6 to 11. In the above formulae,

v_t = pitch line velocity, use equation (1-3-9), ft/min

$A = 50 + 56(1-B)$

$$B = \frac{(12 - Q_v)^{0.667}}{4}$$

Q_v = AGMA gear accuracy grade

$(v_t)_{max} = [A + (Q_v - 3)]^2$, ft/min

From equations (1-4-12), (1-4-13) and (1-3-9), it can be seen that dynamic factors C_v and K_v are function of gear accuracy grade Q_v, pinion pitch circle diameter d and pinion speed n_p.

The load distribution factors C_m and K_m modify the rating equations to reflect the non-uniform distribution of the load over face width. C_m and K_m are defined as the ratio of peak to mean loading. The values of load distribution factors C_m and K_m are related to the precision of the gear box, the face width F and the pinion aspect ratio λ=F/d, as shown in Fig.1-13 and Fig.1-14. Closed form formulae are also given by AGMA as follows.

$$C_m = K_m = 1 + C_{mc}(C_{pf}C_{pm} + C_{ma}C_e) \tag{1-4-14}$$

where

C_{mc} = lead correction factor,

= 1 for gears with unmodified leads

= 0.8 for gears with leads modified

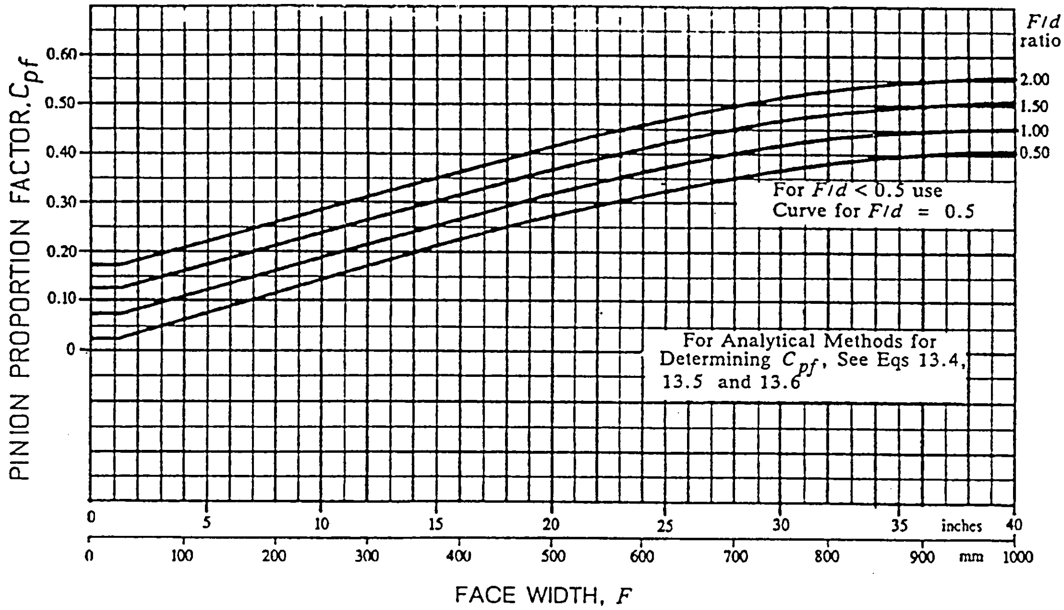


Fig.1-13 Pinion Proportion Factor, C_{pf}

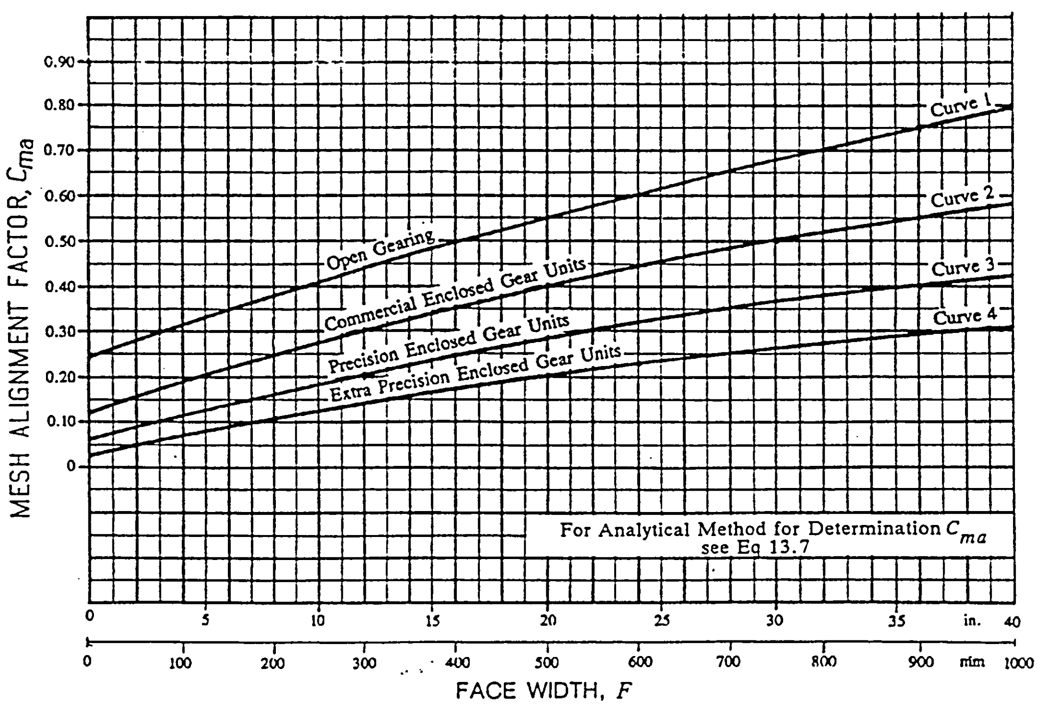


Fig.1-14 Mesh Alignment Factor, C_{ma}

C_{pf} = pinion proportion factor,

$$= \frac{\lambda}{10} - 0.025, \qquad F \leq 1 \text{ inch;}$$

$$= \frac{\lambda}{10} - 0.0375 + 0.0125F, \qquad 1 < F \leq 17 \text{ inch;}$$

$$= \frac{\lambda}{10} - 0.1109 + 0.0207F - 0.000228F^2, \quad 17 < F \leq 40 \text{ inch;}$$

= for values of $\frac{\lambda}{10} < 0.05$, use $\frac{\lambda}{10} = 0.05$ in the above

C_{pm} = pinion proportion modifier,

= 1 for straddle mounted pinions with $(S_1/S) < 0.175$

= 1.1 for straddle mounted pinions with $(S_1/S) \geq 0.175$

where S is the bearing span and

S_1 is the deviation of centre line of gear face from the centre line of bearing span.

C_{ma} = mesh alignment factor,

$= A_1 + A_2F + A_3F^2$

	A_1	A_2	A_3
Open Gearing	0.247	0.0167	-0.765×10^{-4}
Commercial Enclosed	0.127	0.0158	-1.093×10^{-4}
Precision Enclosed	0.0675	0.0128	-0.926×10^{-4}
Extra Precision Enclosed	0.0380	0.0102	-0.822×10^{-4}

C_e = mesh alignment correction factor,

= 0.8 when the gearing is adjusted at assembly or lapped

= 1.0 for other conditions

1.4.3 *Gear Materials and Allowable Stress Numbers*

Although there are a variety of choices for gear materials, such as cast iron, bronze, nylons and plastics, steel is still the predominant material used for power transmission gears. To use the mechanical properties of steel efficiently, steel gears are usually heat treated. The hardness of the steel after heat treatment has a direct influence on the allowable stress numbers for pitting resistance and bending strength. In heavy engineering, through hardened and case hardened gears are most commonly used. *Through hardened* gears are made from medium or high carbon content(0.35% to 0.60%) alloy steels quenched and tempered. The hardness of through hardened gears usually ranges from 180BHN to 400BHN. *Case hardened* gears are made from low carbon content(0.10% to 0.25%) alloy steels with a high carbon content case carburised by carbon containing media and then quenched. The case hardness of case hardened gears usually ranges from 55HRC to 62HRC.

In AGMA 2001-B88, allowable stress numbers for pitting resistance and bending strength s_{ac} and s_{at} are given for unity application factor, 10 million cycles of load application, 99% reliability and unidirectional loading. 70% of the given s_{at} values should be used for idle gears and other gears where the teeth are completely reverse loaded on every cycle. The allowable stress numbers are mainly determined by material and metallurgical quality, type of heat treatment and hardness. The values for s_{ac} and s_{at} for through hardened steels are as shown in Fig.1-15 and Fig.1-16. The values for carburised and case hardened steels are as follows:

AGMA steel:	Grade 1	Grade 2	Grade 3
Min Hardness	55 HRC	58 HRC	58 HRC
s_{ac}(psi)	180000	225000	275000
s_{at}(psi)	55000	65000	75000

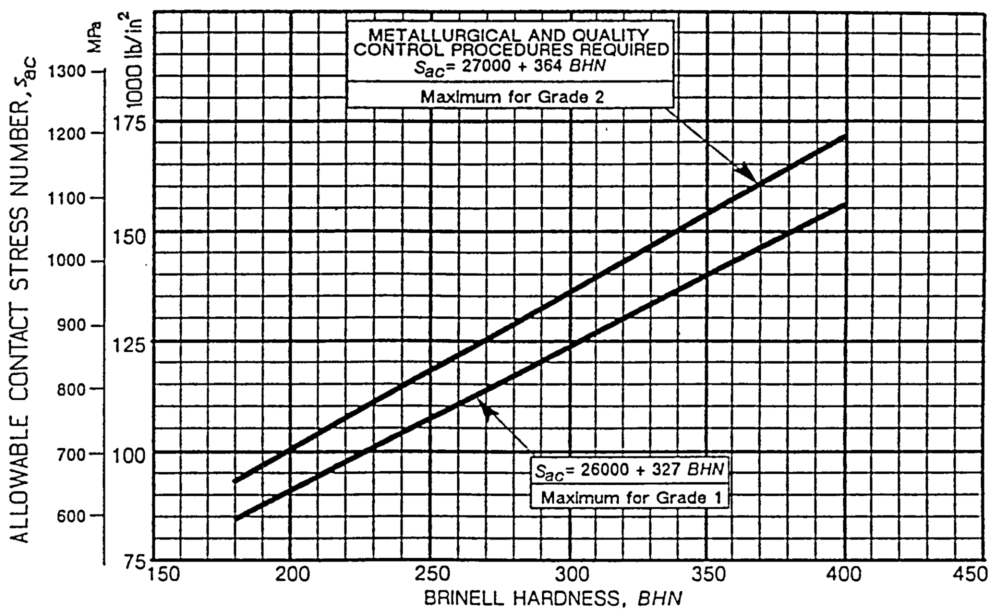


Fig.1-15 Allowable Contact Stress Number for Steel Gears, s_{ac}

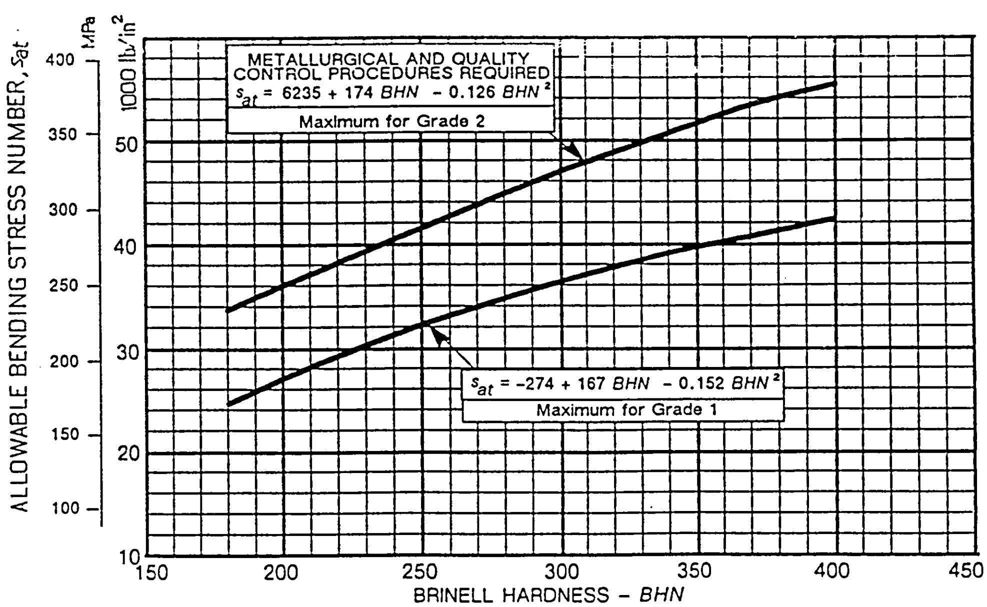


Fig.1-16 Allowable Bending Stress Number for Steel Gears, s_{at}

2.1 *Gear Design by Traditional Methods*

Traditionally, the main dimensions of a gear set, such as centre distance and face width, have been estimated by using some formulae based on pitting resistance. The teeth of through hardened steel gears usually have higher strength to resist bending breakage than they have capacity to resist surface pitting. Thus the pitting resistance capacity becomes the limiting factor in determining the overall size of a gear set of through hardened steels. Representative of this approach is the K factor method as presented by Dudley [28] (1984). The K factor is an index of tooth load intensity pertinent to surface contact stress and is defined as

$$K = \frac{W_t}{d\ F} \left(\frac{m_G + 1}{m_G}\right) \qquad (2\text{-}1\text{-}1)$$

The gear materials must have the capacity to withstand this intensity of loading to be able to function properly. Dudley has compiled a table of allowable values of the K factor for different applications, materials, grades of gear accuracy and load cycles. For convenience of use for design purpose, W_t in equation (2-1-1) is substituted by equation (1-4-3),

$$K = \frac{126000\ P}{n_p\ d^2\ F} \left(\frac{m_G + 1}{m_G}\right) \qquad (2\text{-}1\text{-}2)$$

Then the gear size estimating formula is

$$Fd^2 = \frac{126000\ P}{n_p\ K} \left(\frac{m_G + 1}{m_G}\right) \qquad (2\text{-}1\text{-}3)$$

The formula can also be written as

$$C_r^2F = \frac{31500\ P}{n_p\ K}\ \frac{(m_G + 1)^3}{m_G} \qquad (2\text{-}1\text{-}4)$$

if d in equation (2-1-3) is substituted by equation (1-3-4). Dudley references the quantity $\frac{P(m_G+1)^3}{n_p\ m_G}$ by the letter Q and equation (2-1-4) is simplified to

$$C_r^2F = \frac{31500\ Q}{K} \qquad (2\text{-}1\text{-}5)$$

For a particular gear design problem, the transmitted power P, pinion speed n_p, and gear ratio m_G are usually given. The general procedure given in [28] for sizing a set of gears using the K factor is described as follows. The allowable value of the K factor has to be decided first, according to the application, material, accuracy and load cycles. Equation (2-1-4) or (2-1-5) is then used to estimate the overall size of the gear set, that is, the combination of centre distance and face width. The face width is often related to the pinion pitch circle diameter by a fixed aspect ratio λ. This fixed aspect ratio can be used to eliminate face width F in equation (2-1-4) and equation (2-1-5) so that pinion pitch circle diameter or centre distance can be determined. Face width is calculated by using the pinion aspect ratio afterwards. Once the centre distance and face width have been decided, the size of the teeth is then determined in terms of module or diametral pitch to make the teeth strong enough to take the bending stress in the root of teeth. After the preliminary sizing of the gears, detailed design is done and power capacities are checked by AGMA rating methods.

The idea of estimating the main dimensions of a gear set by pitting resistance has also been exploited by others. MAAG [36](1963) actually used the same method of K factor. However, the values of K factor were taken from Lloyd's Register of Shipping and they are different from Dudley's. The following formula based on pitting resistance was used by Barley [11](1986) to determine the main dimensions of a gear set,

$$Fd^2 = 6.5*10^{11}\left[\frac{P\ C_a\ S_H}{(1.4*HV + 200)^2}\right]\left[\frac{L}{n_p^{7.77}}\right]^{0.114}\left[\frac{G-3}{2.75}\right] \quad (2\text{-}1\text{-}6)$$

where

S_H = minimum factor of safety for pitting resistance

HV = Vicker's hardness number for the gear material

L = expected life in hours

G = gear quality grade to BS 436:part1:1967 [15], or
gear quality grade to BS 436:part2:1970 [16]

The face width F was similarly related to the pitch circle diameter d of pinion by the aspect ratio λ and thus equation (2-1-6) could be solved for d.

The K factor method is a simple way to estimate the centre distance and face width of a gear set. However, comparing equations (2-1-3) and (1-4-5) and noticing equation (1-4-10), it can be seen that the K factor is a combined factor for

$$\frac{C_v}{C_s\ C_m\ C_f\ C_a}\left(\frac{s_{ac}}{C_p}\ \frac{C_L\ C_H}{C_T\ C_R}\right)^2 \quad \text{and} \quad \frac{\cos\phi_r \sin\phi_r}{2\ m_N}$$

This shows that the K factor method only gives a first approximation of the appropriate centre distance and face width. The procedure described above also does not suggest what to do if the rated power capacity is lower than, or much higher than, the required power capacity. Further, when case hardened steels are used, the limiting factor in determining the size of a gear set is often the bending strength. Thus for a pinion pitch circle diameter estimated by the K factor method, even when the minimum number of pinion teeth is used so that a maximum tooth size can be achieved, the power capacity rating of bending strength may still be lower than what is required.

2.2 *Gear Design by Computer Methods*

With the use of computers in industry finding ever wider applications, gear design by computer methods is simply an ideal solution to the problem of lengthy, complex and error prone long hand calculations of gear design. Recently, the K factor method has been computerised by Zarefar and Lawley [53](1989). But one of the earliest papers on the design of gears by computer methods was presented by Cockerham and Waite [21](1976). The method used was a typical cut-and-try method. Minimum number of pinion teeth to avoid involute interference was used to determine module by geometric relation with the centre distance. Face width was then determined by considering pitting resistance and bending strength, using the following equations for power capacities from BS 436:1940 [14], modified for metric unit usage,

$$P_c = \frac{X_c S_c Z F m^{1.8} T N}{1000} \tag{2-2-1}$$

and

$$P_b = \frac{X_b S_b Y F m^{2} T N}{1000} \tag{2-2-2}$$

where

P_c = power for pitting resistance

P_b = power for bending strength

X_c = speed factor for pitting resistance

X_b = speed factor for bending strength

S_c = surface stress factor of pinion or gear

S_b = bending stress factor of pinion or gear

Z = zone factor

Y = strength factor

F = face width

m = module

T = number of teeth of pinion or gear

N = speed of pinion or gear

By applying some constraints on the minimum and maximum face width in relation to pitch circle diameters and module, the face width was either accepted or rejected depending upon whether the constraints on face width were satisfied. If the face width was rejected, a change in material or module was initiated depending upon whether the centre distance was given. This approach was made possible because BS 436:1940 is a relatively simple gear rating standard and the rating factors used in the standard are not related to face width.

Bhattacharjee et al. [12](1986) developed a computer program for the design of spur gears, in which the module of a gear set was determined by static bending strength using the Lewis equation and the minimum number of teeth to avoid involute interference, with centre distance given and face width related to module by a constant. The Lewis equation used to solve for module m was,

$$W_t = SF\pi Ym \tag{2-2-3}$$

where S was allowable static stress and Y was Lewis form factor, being a linear function of the ratio of module to pitch circle diameter. The design was then checked for fatigue bending strength using the Lewis equation with the allowable static stress substituted by endurance stress. Pitting resistance was checked by Buckingham's equations. The equation for the permitted tangential load W_w for pitting resistance was,

$$W_w = 10.13dF\,S_{es}^2 \sin\phi\left[\frac{1}{E_1} + \frac{1}{E_2}\right]\frac{2m_G}{1 + m_G} \tag{2-2-4}$$

where S_{es} is surface endurance limit. The dynamic load W_d calculated by the following equation must be less than the permitted load,

$$W_d = W_t + \frac{0.1105v(Fc + W_t)}{0.1105v + (Fc + W_t)^{1/2}} \tag{2-2-5}$$

where v is pitch line velocity and c is deformation factor. Part drawings of the pinion and gear were created, showing the tooth profile, pitch circle diameter and face width.

A very similar approach was taken by Madhusudan and Vijayasimha in [37](1987). That is, for a given centre distance, the module was determined by static strength and then checked for fatigue strength by the Lewis equation and pitting resistance by Buckingham's equations. The works of [12] and [37] showed one way of determining the module when centre distance is given. However, the formulae used were only crude originals from which the AGMA standards have evolved and are much more complex. The simple relation as shown in equation (2-2-3) is no longer considered sufficiently accurate and thus module m cannot be directly determined.

The work reported in [9](1983) done by Taylor and Walton was a more comprehensive attempt at computerising the design of a gear train system. For the gear design, the standard used was also BS 436-1940 and the same approach was taken as that of [21], i.e., the numbers of teeth were determined by avoidance of involute interference, the gear ratio specified, together with the geometric relation to a given centre distance. The face width was used as the final adjustment variable for satisfying the surface durability and bending strength requirements. Separate programs were written for the design of shafts and selection of bearings. Shaft length, gear and bearing locations, and forces on the shaft were required to determine the shaft diameter by torsional and bending stresses.

Walton et al. [51](1986) gave a more detailed description of the work reported in [9] and further developments, including optimisation of the gear design. Although centre distance, volume and contact ratio were chosen as objectives to be optimised, only the algorithm for minimising the centre distance was shown. The method was based on the understanding that to minimise the centre distance, total number of teeth on the gear pair and module should be as small as possible, whilst face width within practical limits could be adjusted to meet the requirements of pitting resistance and bending strength. Pinion and gear teeth were determined first, then module was increased or decreased depending on whether face width was wider or narrower than the limits. This was repeated until face width was within the limits and module could not be further reduced. One difficulty is that it is not always possible to obtain both minimum teeth and minimum module. Besides, face width as a final adjustment is only possible with BS 436:1940. A graph was shown by the authors for one case indicating that the optimum solution was not given by the design with the minimum number of teeth.

Obviously, the above works were pioneering in using computers for gear designs, but in the light of recent developments, there is a need for new approaches to gear design. Typical of these are new gear rating standards which try to address the problem of gear rating by including many more practical influences and give a more accurate prediction of the power capacity performance of a gear set. For example, all of the modern gear rating standards have factors that are related to face width, especially for load distribution factor calculation, making it impossible to use face width as a final adjustment parameter. AGMA Standards have been widely used in industry and have evolved from decades of R&D and experience of practical engineering use. The rating methods are reasonably accurate and simple, and have proven the most popular.

Computer programs used for refining gear designs according to AGMA Standards were described in [32](1987) by Gitchel. The initial input data to the program for spur and helical gears consist of speed, torque, number of teeth of pinion and gear, normal diametral pitch and normal pressure angle. The program then works out the addendum modifications, operating centre distance and outside diameters, based on considerations of making the AGMA geometry factors J of pinion and gear equal and controlling the sliding ratios. The graphical output of the tooth profiles of pinion and gear helped the user to see the proposed tooth shapes. The program was more for the fine tuning of a roughly given gear set than designing from the basic specification of speed, speed ratio, torque(or power) and materials.

A computer program was developed by Setlur and Andrews [44] (1988) for the design of spur and helical gear pairs, determination of shaft diameters and calculation of bearing loads, of one or two stage gear trains. The gear design started with the calculation for numbers of teeth for both pinion and gear, taking into account of the minimum number of teeth to avoid involute interference and the speed ratio required. Trial values of diametral pitch were then used with the numbers of teeth to determine the pitch circle diameters accordingly. Starting from an assumed value, the trial value of diametral pitch was reduced iteratively in each of three sequential stages so that first, static bending strength, then second, fatigue bending strength, and third, surface pitting resistance, were all satisfied by AGMA rating methods respectively. This approach would certainly result in an adequate design, but not necessarily an optimum design. The question is what number of pinion teeth to use, as this was not really dealt with by the algorithm described in the paper, although the minimum number of teeth was mentioned. A design with the minimum number of pinion teeth is an optimum design only if bending strength is the limiting factor, but not necessarily so if pitting resistance is the limiting factor. As for the part of the package for shaft design, the shaft diameter was determined by static strength and fatigue strength, using maximum shear stress theory. The fatigue bending stress was considered as alternating stress due to the rotation of the shaft and the torsional stress only had a mean value caused by a steady torque. The Soderberg line approach was used for the combination of alternating and mean stresses. Graphic output of gear part drawings was given by AutoCAD using a DXF data file transferred from the design program package.

2.3 *Optimum Design of Single Gear Set*

For most of the earlier works reported, optimisation of the gear set being designed was not considered. The work of [32] (1987) contained an element of optimisation when the J factors of pinion and gear are made equal by adjusting addendum modifications so that equal strength can be obtained for pinion and gear. But this only works when the materials of the pinion and gear are the same. Recent works put more emphasis on the optimum design of gears. In formulating an optimum design problem, an *objective function* must be defined first, which represents a quantity or an index to be minimised or maximised, or in general terms, optimised. The value of the objective function is a function of *design variables* which define or represent a design and can be changed to achieve different designs. The variation of design variables is restricted by design *constraints*, which are equations or inequalities of design variables that must be satisfied. A *feasible design* is a design that satisfies all the design constraints and the set of all the feasible designs is called *feasible domain or region*. The *optimum design* is the feasible design which has the lowest or highest objective function value, depending on whether minimisation or maximisation is desired.

A design procedure for minimising the centre distance of a spur gear set was presented by Savage et al. [40](1982). Face width was related to the pinion pitch circle diameter by the aspect ratio λ. Diametral pitch and number of pinion teeth were the two free design variables. The method was based on the Lewis equation for bending stress and Hertz equation for surface contact stresses for pitting and scoring. The Lewis equation and Hertz equation were used to directly solve for upper bounds on the diametral pitch as a function of number of pinion teeth. Minimum number of teeth to avoid involute interference was also considered. These boundary equations were then used to plot a diagram which showed the feasible region of acceptable designs, with number of pinion teeth and diametral pitch as the two coordinates, as in Fig.2-1. The minimum centre distance was found by drawing a straight line through the origin, at least part of the line being in the feasible region and with a minimum inclination. The straight line represented constant centre distance designs and the minimum inclination meant minimum centre distance. Because

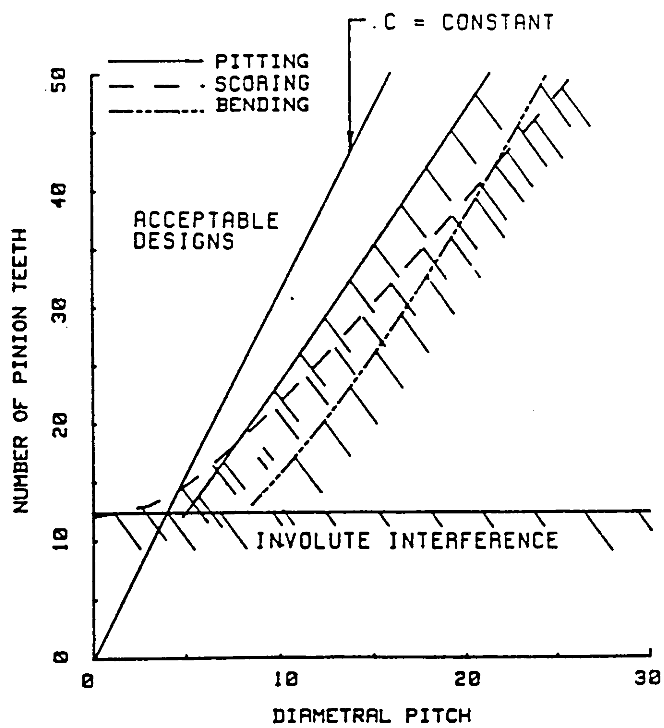


Fig.2-1 Feasible Region and Minimum Centre Distance

the procedure used a graphic method, the effect of design constraints on centre distance was very clear. The difficulty was probably in constructing the diagram for each design job and computerising the procedure. The simplifications used by the authors included the use of Lewis form factor Y instead of AGMA geometry factor J. Dynamic load and non-uniform load distribution across the face were also not considered. Without these simplifications, the procedure would not have been possible.

Carroll and Johnson [20](1984) expanded the work reported by Savage et al. by including AGMA geometry factor J and dynamic factor C_v and K_v. The objective was to minimise the pitch circle diameter of pinion which was equivalent to minimising the centre distance. Face width was similarly related to the pitch circle diameter of pinion. Diametral pitch and number of pinion teeth were the variables. Because of the complexity introduced by the inclusion of geometry factor and dynamic factor, a numerical procedure was proposed to solve the problem. The authors reported difficulties in using general purpose optimisation algorithms such as steepest descent and conjugate gradient methods, that the algorithms stalled prematurely at improved but less than satisfactory designs. Hence a special purpose algorithm was developed. The basic idea was that to minimise the pitch circle diameter of pinion, number of pinion teeth must be minimised for a given diametral pitch. The algorithm was thus a one dimension or line search technique for determining minimum number of pinion teeth for each of a range of candidate diametral pitches. The technique used was one that keeps doubling step size to determine the bracket containing the minimum and then keeps halving step size to locate the minimum. It required a feasible initial design to start the line search. The implicit assumption used by the authors was that if a design with a certain number of pinion teeth could not satisfy all the constraints, any design with a smaller number of pinion teeth would not satisfy the constraints. The final optimum design was found by simple comparison of the minimum centre distances found for all the diametral pitches.

Vanderplaats et al. [50](1988) reported using a general purpose optimisation program for the optimum design of gear sets. The general purpose optimisation program was a stand alone, independent program containing a variety of optimisation algorithms. An analysis program must be supplied by the user to define the objective, the constraints and the variables of the optimum design problem. The optimisation program and the user supplied analysis program were then linked by a control program which acted as the interface between the two. The method was actually applied to maximise pitting resistance life and minimise dynamic load of spur and spiral bevel gears, respectively. The development of the gear analysis program was shown to include constraints on pitting resistance, bending strength and scoring conditions. Design variables were numbers of teeth of pinion and gear, diametral pitch, and face width. Obviously, the solution of the optimum gear design problems depends a lot on the optimisation algorithms used. However, little was given about how the algorithms worked and which were more effective and efficient in solving the optimum gear design problems.

For the optimum design of helical gears, Jog and Pande [34] (1989) used an interior penalty function method combined with Powell's conjugate direction method [39](1984) to minimise the volume of the gear set, considering bending strength, pitting resistance and avoidance of involute interference. Free design variables were number of pinion teeth and helix angle. The optimisation was done for each of a range of candidate diametral pitches. The addendum modifications were determined by avoidance of involute interference and balancing of bending strengths of pinion and gear. The face width was a dependent variable defined by

$$F = \frac{m_F \pi m_n}{\sin\psi} \qquad (2\text{-}3\text{-}1)$$

to obtain the required face contact(overlap) ratio m_F. Helix angle was discretised to increase the computational speed and reduce storage. The idea of the penalty function method is to introduce into the original objective function some penalty items related to the constraints so that the penalty items become increasingly larger when a design approaches the constraints, preventing the violation of constraints while minimising the objective function using some unconstrained optimisation method such as that of Powell's. Because optimum gear designs are generally at or near the constraints of pitting resistance or bending strength, the penalty function can increase rapidly and only a near optimum solution can be found by the method. The efficiency or convergence speed of the method is sensitive to the starting solution and some parameters controlling how the penalty items should be increased. The authors gave a method on how the starting solution should be determined and suggested some values for the controlling parameters.

Zarefar and Muthukrishnan [54](1992) reported using a random search method for the minimisation of the weight of a gear set. Pitting resistance and bending strength were considered as design constraints. Module, helix angle, number of pinion teeth and face width were the design variables. The algorithm started from a starting solution and searched along a number of randomly generated directions. In each of the random directions, a number of solutions were created by changing the step size of search. If any constraint was violated, the direction of search was reversed. The number of random directions and the number of searches in each direction were determined by the user. The strategy used in the random search algorithm was to generate as many feasible solutions as possible and to determine the best solution from the set of feasible solutions.

A two stage optimisation approach was employed by Prayoonrat and Walton [38](1988) to minimise the centre distance of helical gears in terms of module and numbers of teeth of pinion and gear. Helix angle was to be specified by the user. In the first stage, a direct search method was used and the requirement that numbers of teeth and module must be discrete values was ignored. In the second stage, the results from the first stage were rounded and the effects of small changes in numbers of teeth and module were examined, which the authors called a heuristic method. BS 436:1940 was used and face width was adjusted to satisfy pitting resistance and bending strength. When the face width so determined was thinner than the lower limit, the lower limit was used. If the face width was wider than the upper limit, a partial penalty method was used, i.e., the difference was squared and added to the quantity being minimised which was centre distance. The direct search method was a sequence of unidirectional searches followed by a rotation of the search directions as proposed by Davies, Swann, and Compey [13] (1969). Difficulties were reported when setting the convergence criteria for the direct search method, that the iteration could either terminate prematurely or be trapped in an endless loop. To overcome these difficulties, the authors suggested some minimum and maximum numbers of iterations, based on their computational experience. Number of iterations in the heuristic method depended on how much the user wanted to widen up the search after the first stage.

Errichello [29](1989) proposed a closed-form procedure for the optimum design of spur and helical gears. The method was based on the idea of an optimum number of pinion teeth which would result in balanced pitting resistance and bending strength. The procedure started by solving for the pitch circle diameter of pinion based on pitting resistance, similar to that of the K factor method, with face width related to pinion diameter by the aspect ratio. The optimum number of pinion teeth was then solved for by simultaneously satisfying the pitting resistance and bending strength equations. To develop the closed-form equations to solve for the pitch circle diameter of pinion and the optimum number of pinion teeth, the author had to approximate the AGMA geometry factors I and J, dynamic factor C_v and K_v, and load distribution factor C_m and K_m, which meant the optimum design must be verified by the relevant AGMA standard afterwards.

2.4 *Optimum Design of Gear Systems*

For multi-reduction gear trains, how to split the overall gear ratio into ratios of each reduction is another problem in gear design that needs to be addressed. Willis [52](1963) developed a set of equations for splitting the overall gear ratio of different gear systems such that minimum weight gears could result. The method was based on the solid rotor volume of the gears and Dudley's K factor method, with an assumption that the materials used for all the gears were the same. With the solid rotor volume of a pinion being $V=\pi d^2F/4$, the right hand side of equation (2-1-3) was used to calculate the solid rotor volume of the pinion. Similar relations were developed for other gears in a gear system. The volume of a gear system, such as that of a double reduction gear train, was finally expressed in the form of a volume index as a function of overall gear ratio m_O and gear ratio m_{G1} of the first reduction, while transmitted power P, pinion speed n_p and K factor were considered known and combined into a constant in the volume index. The volume index of double reduction gear train was given as

$$1 + \frac{1}{m_{G1}} + 2m_{G1} + m_{G1}^2 + \frac{m_{G1}^2}{m_O} + \frac{m_O^2}{m_{G1}} + m_O \qquad (2\text{-}4\text{-}1)$$

The minimum of the function of volume index was found by analytical method with the gear ratio m_{G1} of the first reduction treated as a variable. The equation that the gear ratio m_{G1} had to satisfy to achieve minimum volume design for a double reduction gear train was then derived as

$$2m_{G1}^3 + \frac{2m_{G1}^2}{\left(\frac{m_O + 1}{m_O}\right)} = \frac{m_O^2 + 1}{\left(\frac{m_O + 1}{m_O}\right)} \qquad (2\text{-}4\text{-}2)$$

Similar equations for volume index and minimum weight gear ratio were given for the eight gear systems as shown in Fig.2-2. Curves for minimum weight gear ratio were also given

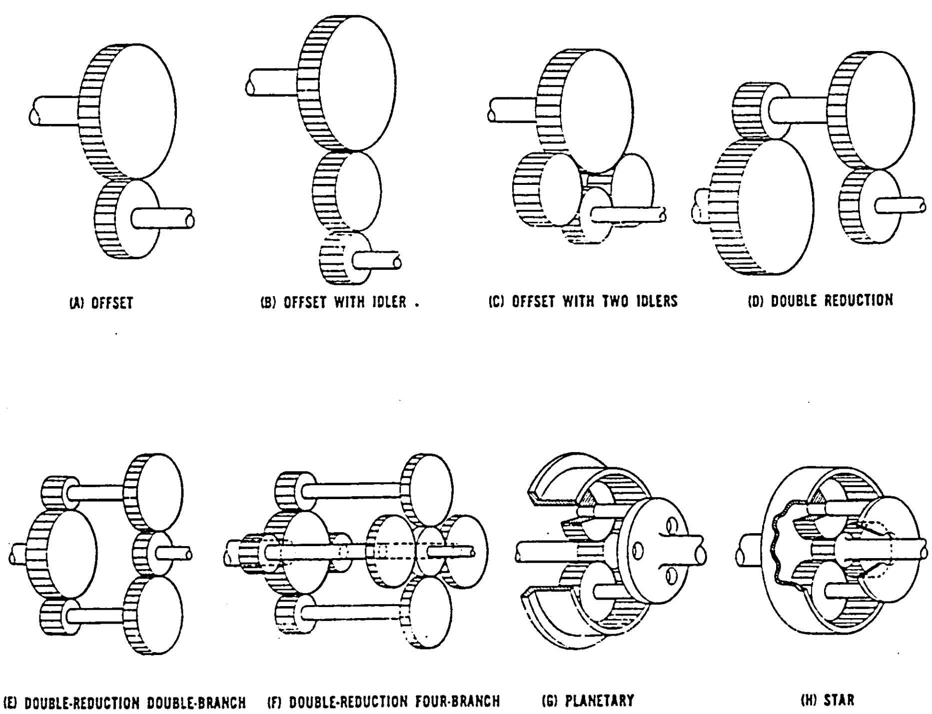


Fig.2-2 Eight Gear Systems

to facilitate the selection of a value for the gear ratio, since equations like (2-4-2) were difficult to solve for a value of m_{G1} without resorting to numerical methods. ESDU 88033 [31](1988) gave the following simpler formulae for the division of overall gear ratio for double and triple reduction gear boxes which are derived by Niemann and Winter to achieve the minimum total gear volume.

$$m_{G1} = 0.8m_o^{2/3} \tag{2-4-3}$$

for double reduction gear box, and

$$m_{G1} = 0.6m_o^{4/7} \tag{2-4-4}$$

$$m_{G2} = 1.1m_o^{2/7} \tag{2-4-5}$$

for triple reduction gear box, where m_{G1} and m_{G2} are the gear ratios for first and second reductions, respectively. ESDU commented that although achieving minimum total gear volume, these ratio splits result in high ratios in the first stages, which may not represent the most cost-effective ratio split. Consequently, the formulae should be used for guidance only.

An attempt was made by Abdelhafez [1](1981) to develop a computer program package for the design of mechanical power transmission systems with multiple outputs and power paths, constructed from different types of drive units, including belt, chain and gear drives. The concepts of node, path and flow graph as from graph theory were used to describe any such system. Matrices and Cartesian coordinate systems were then used to represent the logical power paths, geometrical relations and functional requirements of the different drive units. Each drive unit on each power path was designed individually. No information was given on how these drive units were designed by computer methods. The use of graph theory is a systematic and clear way for the representation of drive systems of complex layout with different drive types and multiple outputs and power paths.

Mathematical models for optimum design of gear transmission system were presented by Shchekin [45](1987). The gear transmission system was conceptually divided into five modules for the logical organisation of design data and algorithms. These five modules were gear tooth, gear blank, shaft, bearing and layout. For the optimum design of a gear system, the author proposed an objective function which was the weighted sum of contact stress and bending stress. For the optimum selection of addendum modifications, minimisation of either contact stresses or bending stresses was proposed as the objective. Only conceptual formulations were discussed. No design method or procedure was given to solve the problems.

Savage et al. [41](1992) applied optimisation to the design of a single reduction spur gear transmission system, including gears, bearings and shafts. The objective was to design a small, light weight transmission system with a long service life. The objective function was formulated to maximise service life with penalty for large volume and heavy weight. Bearing life and gear pitting resistance life were considered for the service life. Different penalties for volume and weight were experimented and the resulting solutions showed shifts in the relative importance of life, volume and weight. Design variables included diametral pitch, face width, number of pinion teeth, axial locations of the bearings and the two shaft diameters. Design constraints considered were: gear stresses, involute interference, gear rim thickness, shaft stiffness, axial clearance between gear and bearing, and radial clearance between bearing O/D's. A general purpose optimisation algorithm, modified feasible direction gradient method was used to solve the optimisation problem. Basically, the method is a gradient(steepest descent) method. When the design violates or is near the constraints, the steepest decent direction is modified by a direction pointing to the feasible region, thus preventing violation of constraints. Polynomial interpolating functions were used to represent the discrete catalogue bearing data to facilitate the selection of bearings. The designs obtained by the gear optimisation and bearing selection were continuous variable solutions which must be modified by the user to conform to the discrete requirements.

2.5 *Optimum Selection of Addendum Modification*

Buchhorn et al. [19](1981) addressed another problem in the optimum design of spur and helical gears, the optimum tooth shape as affected by addendum modification. The standard the authors used as the basis for the analysis was Australian Standard AS B61 which is the same as BS 436:1940. Results for positive addendum modification on pinion(x_1=0.32) and negative on gear(x_2=-0.32) were compared with results for positive addendum modifications on both pinion(x_1=0.70) and gear (x_2=0.84). The conclusion reached was that positive addendum modifications on both pinion and gear would result in higher values for the strength factor Y and Zone factor Z, and hence higher capacity ratings for transmitted torque or power.

The same topic was discussed by Andrews and Argent [2] (1992) as to how addendum modification would affect the performance of a gear set. A slightly different viewpoint was taken that the optimum addendum modifications should make equal the AGMA geometry factors J for bending strength of pinion and gear. The assumption was that equal geometry factors would result in equal strength, which in turn would result in optimum strength of the gear set since the strength of the gear set was determined by the weaker member of the pair. The method was for modifying known or existing gear sets with fixed centre distances. Two cases were considered for the addendum modifications. In one case, positive and negative addendum modifications of the same amount were applied to pinion and gear respectively. In the other case, one tooth was 'dropped' from the pinion or the gear and the addendum modifications were used to make up the difference in centre distance caused by the 'dropped' tooth. The method was reported to be efficient in obtaining equal J factors in a few iterations of the optimisation process.

The topic of addendum modification has also been covered by others. The general opinions of BSI PD 6457 [18], DIN 3992 [26], ESDU 77002 [30], MAAG [36], Dudley [28] and Errichelo [29] are: addendum modifications should be a compromise between the balancing of bending strengths and the balancing of sliding ratios, while consideration is also given to the avoidance of undercut and narrow tooth top land. Positive sum of addendum modification coefficients Σx leads to a larger working pressure angle because of the extended centre distance and unchanged base circle diameter. The larger pressure angle is good for both strength and durability but reduces transverse contact ratio, making the gearing noisier. The benefit that is gained from stronger tooth root and larger radius of curvature may be offset by the reduced contact ratio. Negative Σx leads to a smaller working pressure angle and larger transverse contact ratio, resulting in quieter gears but not as strong. For balanced designs, DIN proposed a range of values between 0.0 and 0.6, while BS recommended Σx should be kept below 0.4. Dudley advocated $\Sigma x=0$ design with positive addendum modification on pinion and negative on gear.

For the division of addendum modifications between pinion and gear, BSI [18](1970) gives the following equations for three different applications:

1.For general application(recess action slightly longer than approach action):

$$x_1 = \frac{1}{3}\left(1 - \frac{1}{m_G}\right) + \frac{\Sigma x}{1 + m_G} \qquad (2\text{-}5\text{-}1)$$

2.For approximate equality of bending strength factors for pinion and gear:

$$x_1 = \frac{1}{2}\left(1 - \frac{1}{m_G}\right) + \frac{\Sigma x}{1 + m_G} \qquad (2\text{-}5\text{-}2)$$

3. For approximate balance of worst sliding ratios at pinion and gear tips:

$$x_1 = \frac{2}{(N_1/\cos^3\psi)^{1/2}}\left(1 - \frac{1}{m_G}\right) + \frac{\Sigma x}{1 + m_G} \qquad (2\text{-}5\text{-}3)$$

A diagram as shown in Fig.2-3 was given in DIN [26](1964) for the division of addendum modifications. The average number of teeth $(N_1+N_2)/2$ and average addendum modification coefficient $(x_1+x_2)/2$ are used to locate a point in the diagram. A line passing through the point and having a similar slope to the neighbouring lines is drawn. Using N_1 and N_2, x_1 and x_2 can be found on the line. The diagram is based on a compromise that the bending strengths of pinion and gear be balanced, pinion tip sliding velocity be slightly greater than gear tip sliding velocity and that extreme sliding ratios be avoided. It can be seen that lines for avoiding undercut, narrow top land and low transverse contact ratio are also drawn on the diagram.

Fig.2-4 is a diagram given by Dudley [28](1984) for the selection of addendum ratio. The curves in the diagram were drawn to give an approximate balance between the strengths of pinion and gear. It also took care of the problem of undercut. The problem of a large addendum modification was commented as pinion being substantially stronger than gear and scoring at pinion tip becoming more likely.

MAAG [36](1963) gave similar diagrams to those given by DIN [26] for the selection and division of addendum modifications.

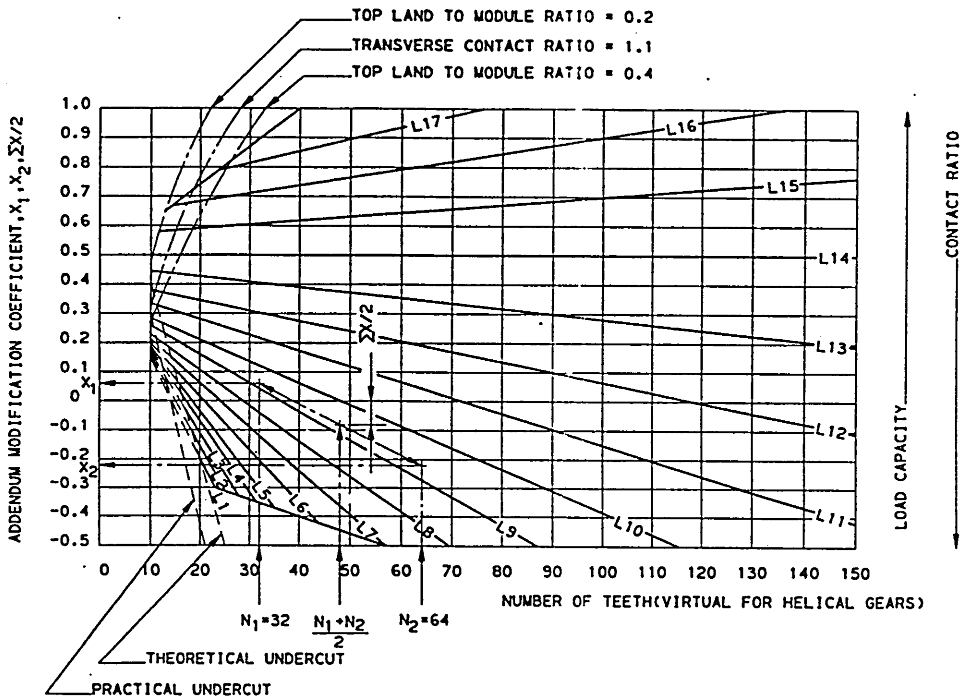


Fig.2-3 Division of Addendum Modification by DIN

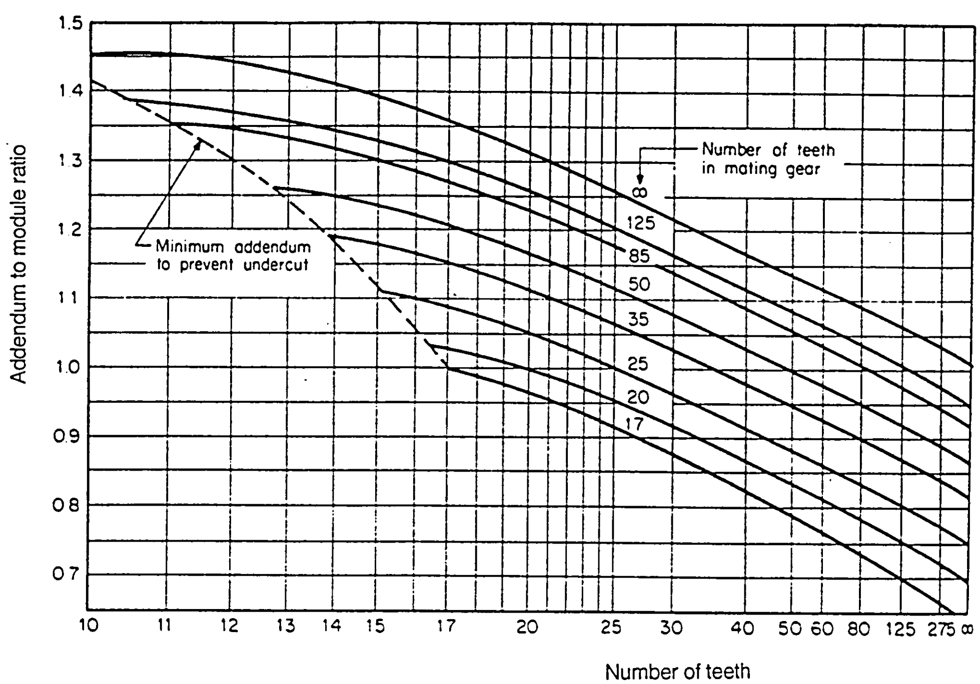


Fig.2-4 Addendum Modification Recommended by Dudley
$\Sigma x=0$

However, MAAG also gave a set of diagrams for controlling the meshing conditions based on the following empirical rules:

1. sliding ratio at the pinion tip should not exceed that at the gear tip;
2. sliding ratio at the gear tip(pinion root) should be less than 3;
3. approach action should be less than recess action;
4. the active profile length of the gear should not be less than 3/4 of the active profile length of pinion.

The diagrams are as shown in Fig.2-5 and are given in terms of tooth contact parameters which are defined as

$$u_1 = \frac{m_G + 1}{m_G}\left(1 - \frac{\tan\phi_r}{\tan\phi_{a1}}\right) \tag{2-5-4}$$

for pinion, and

$$u_2 = (m_G + 1)\left(1 - \frac{\tan\phi_r}{\tan\phi_{a2}}\right) \tag{2-5-5}$$

for gear, where ϕ_{a1} and ϕ_{a2} are pressure angles at the pinion tip and gear tip, respectively. The sliding ratio at the pinion tip is then expressed as

$$\gamma_1 = \frac{u_1}{1 - u_1} \tag{2-5-6}$$

and that at the gear tip is

$$\gamma_2 = \frac{u_2}{1 - u_2} \tag{2-5-7}$$

Obviously, from equations (2-5-6) and (2-5-7), balanced tooth contact parameters u_1 and u_2 result in balanced sliding ratios γ_1 and γ_2 at the tips of pinion and gear. According to MAAG's recommendations, gear sets should be designed to fall within the shaded area in the diagrams as far as possible. It was also noted by MAAG that it is apparent that for gears whose sliding conditions are balanced, it is always possible to obtain a reasonable ratio between the load capacities of pinion and gear teeth.

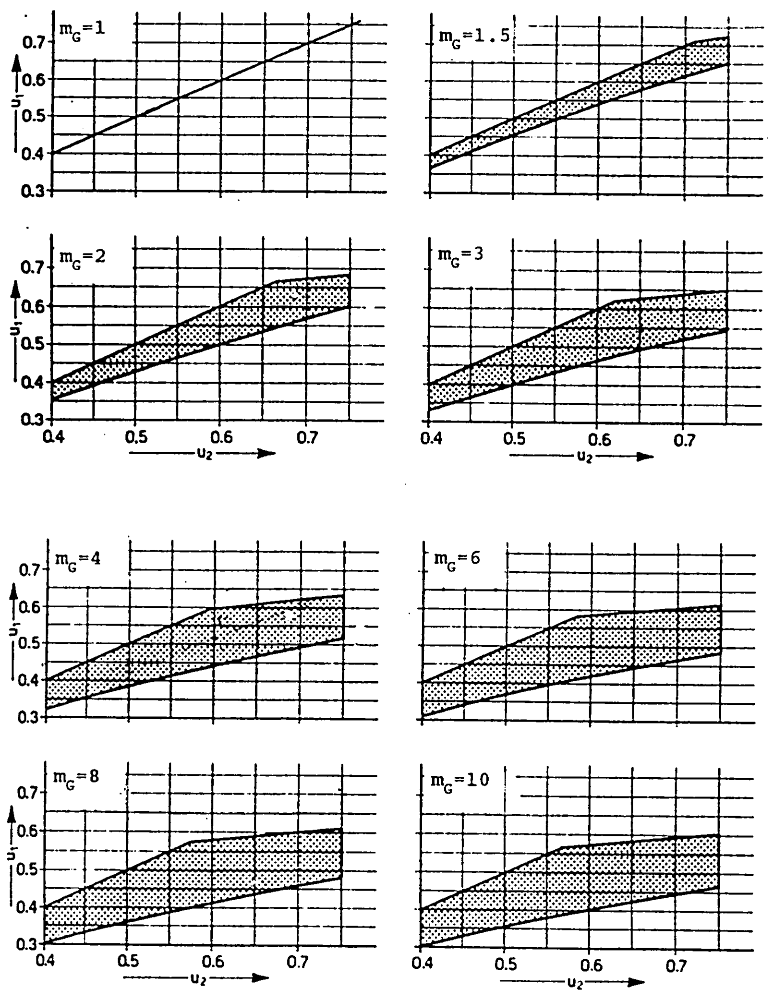


Fig.2-5 Meshing Conditions by MAAG Tooth Contact Parameter u_1 for pinion and u_2 for gear, for different gear ratios(m_G)

2.6 *Summary*

Gear design has been an old art. In recent times, it is becoming a science. There has been a wealth of literature published on the subject of gear design and the literature surveyed above is a representative part.

To summarise,

1. In determining size and geometry for power transmission gears, pitting resistance and bending strength are the two most important influential factors. Pitting resistance and bending strength are often treated as design constraints. Different standards give different formulae and AGMA has proven to be a practical, relatively simple and accurate, and the most popular standard.
2. The traditional way of gear design is to use pitting resistance to determine the centre distance and face width, and then use the bending strength to determine tooth size, i.e., module or diametral pitch. This may not always work if the deciding factor is the bending strength.
3. Earlier works on computerising gear design used the iterative nature of computer algorithms to find an adequate design which would satisfy pitting resistance and bending strength. The general approach is to determine the number of pinion teeth by avoidance of involute interference and number of gear teeth by gear ratio. The tooth size is increased from a small value until pitting resistance and bending strength are satisfied. This results in an adequate design but not necessarily an optimum design.

4. For the optimum design of gears, weight, volume or centre distance have been the most commonly chosen objectives to be minimised, since weight, volume or centre distance are the main indices of the size of a gear box. For a constant material density and fixed pinion aspect ratio, the volume, weight, and centre distance are related to each other by constants and minimisation of any one of them leads to minimisation of the others.
5. Face width has been used in several papers as the final adjustment variable for satisfying the bending strength and pitting resistance, which was only possible with BS 436:1940. A convenient and more practical way is to use the aspect ratio of face width to pinion pitch circle diameter. This ratio has effects on load distribution across the face and gear wheel stability.
6. For the optimum design of gears, a number of approaches have been taken which vary between two extremes: general purpose algorithm versus special algorithm with each having its merit and weakness. Representative of the general purpose algorithm approach are [50], [34] and [54]. The advantage of using a general purpose algorithm for the optimum design of gears is that any change in the design analysis model does not necessitate change of the optimisation algorithm. For the purpose of research, to compare the optimum design results of different models, e.g., changing the objective from minimising centre distance to maximising life, a general purpose algorithm is more suitable. However, it has been well known that there

is no single algorithm which is universally applicable to any optimisation problem and general purpose algorithms can only guarantee a local minimum and do not always terminate at the global optimum. Convergence speed can also be slow. It is not uncommon for a general purpose algorithm to call and evaluate the analysis model thousands of times in the optimisation process, which is time consuming for a gear rating model. The weakness of the general purpose algorithm approach is in its effectiveness and efficiency. In contrast to this, a special purpose algorithm is designed for a particular type of model, it is more effective and efficient because inherent relations in the model can be exploited in the algorithm. Methods described in [40], [20] and [29] can be grouped under the special purpose algorithm category. The weakness in this approach is that the algorithm is only applicable to one type of optimum design model. However, for the specific problem of minimising weight, volume or centre distance of a gear set, an effective and efficient method is all that matters. For the purpose of a daily optimum design tool for gears, special purpose algorithm is more suitable, which can give reliable optimum designs quickly. Use of inherent relations helped to simplify the algorithms given in [40], [20] and [29]. However, the algorithm given in [40] is a graphical procedure that is for spur gears only. [20] also only dealt with spur gears and number of pinion teeth was the only design variable. The algorithm proposed in [29] uses approximate values for AGMA factors which means the result obtained may need to be modified and not be an optimum design any more.

7. The division of overall gear ratio for multi-stage reduction gear box has been approached from the standpoint of minimising weight or volume and relatively simple formulae have been obtained [52] and [31]. The design of a gear transmission system [41] has an added constraint of avoiding physical interference or closeness between bearing O/D's. Shaft deflection is another constraint that needs to be considered. The layout representation is another problem to be solved in the design of a transmission system and [1] attacked this problem by using graph theory and Cartesian coordinate system.

8. Addendum modifications can improve the tooth shape and thus achieve better pitting resistance and bending strength. The difficulty is to strike a balance between avoidance of involute interference, avoidance of narrow top land, balanced strength and balanced sliding ratio. These requirements are not always consistent and can be in conflict. Gearing with positive addendum modification on pinion and an equally large, negative addendum modification on gear is to be favoured. This kind of gearing is simple to design and manufacture, with a standard centre distance and standard pressure angle, while at the same time can take advantage of improved tooth shape. It is apparent that addendum modifications should be determined mainly by balancing of sliding ratios, with the involute interference and narrow top land conditions checked, while a reasonable

balance between the strengths of pinion and gear can be achieved.

9. Graphical output has only been included for component drawing and tooth profile, showing only the gears but not the gear transmission system which also includes bearings, shafts and their arrangement.

To conclude: there is a need to develop an effective and efficient algorithm for the optimum design of helical gears. This may be accomplished by exploiting inherent relations in gear design and using an unsimplified modern and up-to-date standard, such as AGMA 2001-B88, so that the designs obtained can be directly used. In addition, all the supporting shafts and bearings should be included in an automatically generated parametric layout.

3.1 *General Formulation of Optimum Design Problem*

The general form of the mathematical model of an optimum design problem can be represented as follows [39]:

find

$$X = (x_1, x_2, \ldots, x_n) \tag{3-1-1}$$

which minimises

$$f(X) = f(x_1, x_2, \ldots, x_n) \tag{3-1-2}$$

subject to

$$g_i(X) \le 0, \qquad i=1,2,\ldots,k \tag{3-1-3}$$

$$h_i(X) = 0, \qquad i=(k+1),(k+2),\ldots,m \tag{3-1-4}$$

where X is a vector representing n design variables, f(X) is the objective function to be minimised, $g_i(X)$ are the inequality constraints and $h_i(X)$ are the equality constraints that a design is subject to. The objective function f(X) and the constraint functions $g_i(X)$ and $h_i(X)$ are often collectively referred to as the analysis model, while the method used to find the X which minimises f(X) is called algorithm. Note that the above presentation does not lose its generality when the objective function is to be maximised or the inequality constraints are required to be greater than or equal to zero since maximising f(X) is equivalent to minimising -f(X) and $g_i(X) \ge 0$ is equivalent to $-g_i(X) \le 0$.

3.2 *Formulation of Optimum Gear Design Problem*

When designing gear sets for power transmission the minimisation of centre distance is a common requirement, especially if face width is in proportion to pinion pitch circle diameter. The validity of centre distance as the objective function is based upon it being the major index of the size of a gear box. By fixing the aspect ratio of face width to pinion pitch circle diameter, minimisation of centre distance can generally lead to a smaller size gear box when at the same time a good proportion can be maintained for the gear blank. As pointed out in the summary of Chapter Two, minimisation of centre distance with fixed pinion aspect ratio is equivalent to minimisation of weight or volume of the gear set. The optimum gear design problem to be solved can be descriptively defined as follows:

1. The objective is to minimise centre distance C;
2. Design constraints which must be satisfied include power capacities for pitting resistance P_{ac} and for bending strength P_{at}, maximum helix angle ψ_{max}, minimum face contact ratio $(m_F)_{min}$, minimum number of pinion teeth $(N_1)_{min}$, gear ratio tolerance δ, zero sum of addendum modifications Σx, and a discrete set of modules;
3. Independent design variables are the normal module m_n, number of pinion teeth N_1, number of gear teeth N_2, and helix angle ψ;
4. Fixed design parameters include standard normal pressure angle ϕ_n, addendum to module ratio h_a, dedendum to module ratio h_b, fillet radius to module ratio r_b, pinion aspect ratio λ, material properties C_p, s_{ac} and s_{at}, and AGMA gear accuracy grade Q_v;

5. Design specifications include the power to be transmitted P, input pinion speed n_p, required gear ratio m_G, and application of the gear set, C_a and K_a.

Following the general formulation of optimum design problem, the above optimum gear design problem can be expressed mathematically as follows:

find

$$X=(m_n, N_1, N_2, \psi) \tag{3-2-1}$$

which minimises

$$f(X) = C = \frac{(N_1 + N_2)m_n}{2\cos\psi} \tag{3-2-2}$$

subject to

$$g_1(X) = P - P_{ac1} \leq 0 \tag{3-2-3}$$

$$g_2(X) = P - P_{ac2} \leq 0 \tag{3-2-4}$$

$$g_3(X) = P - P_{at1} \leq 0 \tag{3-2-5}$$

$$g_4(X) = P - P_{at2} \leq 0 \tag{3-2-6}$$

$$g_5(X) = \psi - \psi_{max} \leq 0 \tag{3-2-7}$$

$$g_6(X) = (m_F)_{min} - m_F \leq 0 \tag{3-2-8}$$

$$g_7(X) = (N_1)_{min} - N_1 \leq 0 \tag{3-2-9}$$

$$g_8(X) = \frac{N_2}{N_1} - (1+\delta)m_G \leq 0 \tag{3-2-10}$$

$$g_9(X) = (1-\delta)m_G - \frac{N_2}{N_1} \leq 0 \tag{3-2-11}$$

$$h_1(X) = P_{ac1} - \frac{n_p F}{126000} \frac{I C_v}{C_s C_m C_f C_a} \left(\frac{d s_{ac1}}{C_p} \frac{C_{L1}}{C_T C_R}\right)^2 = 0 \tag{3-2-12}$$

$$h_2(X) = P_{ac2} - \frac{n_p F}{126000} \frac{I C_v}{C_s C_m C_f C_a} \left(\frac{d s_{ac2}}{C_p} \frac{C_{L2} C_H}{C_T C_R}\right)^2 = 0 \tag{3-2-13}$$

$$h_3(X) = P_{at1} - \frac{n_p\ F}{126000}\ \frac{d}{P_d}\ \frac{J_1\ K_v}{K_s\ K_m\ K_{B1}\ K_a}\ \frac{s_{at1}\ K_{L1}}{K_R\ K_T} = 0 \qquad (3\text{-}2\text{-}14)$$

$$h_4(X) = P_{at2} - \frac{n_p\ F}{126000}\ \frac{d}{P_d}\ \frac{J_2\ K_v}{K_s\ K_m\ K_{B2}\ K_a}\ \frac{s_{at2}\ K_{L2}}{K_R\ K_T} = 0 \qquad (3\text{-}2\text{-}15)$$

$$h_5(X) = x_1 + x_2 = 0 \qquad (3\text{-}2\text{-}16)$$

$$h_6(X) = d - \frac{N_1 m_n}{\cos\psi} = 0 \qquad (3\text{-}2\text{-}17)$$

$$h_7(X) = P_d - \frac{\cos\psi}{m_n} = 0 \qquad (3\text{-}2\text{-}18)$$

$$h_8(X) = F - \lambda d = 0 \qquad (3\text{-}2\text{-}19)$$

$$h_9(X) = m_F - \frac{F\sin\psi}{\pi m_n} = 0 \qquad (3\text{-}2\text{-}20)$$

The equations required to evaluate the items contained in equations (3-2-12) to (3-2-15), such as (1-4-10) for I, (1-4-11) for J, (1-4-13) for C_v and K_v, and (1-4-14) for C_m and K_m, can be considered to be the additional equality constraints to the above formulation. Since the application factors C_a and K_a, material allowables s_{ac}/C_p and s_{at}, life factors C_L and K_L, reliability factors C_R and K_R, temperature factors C_T and K_T, hardness ratio factor C_H, size factor C_s and K_s, surface finish factor C_f, and rim thickness factor K_B, are not related to the design variables $X=(m_n, N_1, N_2, \psi)$, equations (3-2-12) to (3-2-15) can be expressed in simplified forms by amalgamating these factors into C_w and K_w:

$$P_{ac1} - C_{w1}\ n_p \frac{C_v}{C_m}\ I\ F\ d^2 = 0 \qquad (3\text{-}2\text{-}12a)$$

$$P_{ac2} - C_{w2}\, n_p \frac{C_v}{C_m} I\, F\, d^2 = 0 \tag{3-2-13a}$$

$$P_{at1} - K_{w1}\, n_p \frac{K_v}{K_m} J_1\, F \frac{d}{P_d} = 0 \tag{3-2-14a}$$

$$P_{at2} - K_{w2}\, n_p \frac{K_v}{K_m} J_2\, F \frac{d}{P_d} = 0 \tag{3-2-15a}$$

where

$$C_w = \frac{1}{126000} \frac{1}{C_s\, C_f\, C_a} \left(\frac{s_{ac}}{C_p} \frac{C_L\, C_H}{C_T\, C_R}\right)^2$$

is the amalgamated factor for pitting resistance and

$$K_w = \frac{1}{126000} \frac{1}{K_s\, K_B\, K_a} \frac{s_{at}\, K_L}{K_R\, K_T}$$

is the amalgamated factor for bending strength.

3.3 *Explanation of Optimum Gear Design Model*

To study the functional relationships between the terms in the above formulation, they can be divided into independent design variables, dependent variables, boundary limits, fixed parameters and design specifications. All the terms, except for the dependent and independent design variables, are either to use recommended values or to be given values by the design engineer or the customer. From a mathematical point of view, boundary limits, fixed parameters and design specifications are all fixed terms and they can be considered as known in the above formulation and optimisation process.

Independent design variables are terms that can be actively changed to obtain different designs. A design in the context of the above formulation is a particular combination of (m_n, N_1, N_2, ψ). It is obvious that N_1 and N_2, numbers of teeth of pinion and gear, can only take positive whole numbers. Because of standardised gear cutting tools, normal module m_n can only take values from a discrete set. Although normal module m_n is usually given in mm, m_n is expressed in inches in this thesis for consistency of units in AGMA formulae. Thus for the independent design variables, only helix angle ψ can take continuous values. This means that the gear optimum design problem is a mixed discrete variable problem.

Dependent variables are terms whose values are determined by the values of other terms in the formulation. Such variables include C, F, d, P_d, P_{ac}, P_{at}, m_F, I, J, C_v, K_v, C_m, K_m, x_1, and x_2. The equations for the calculation of I, J, C_v, K_v, C_m, K_m are not directly given in the above formulation, because they involve further equations for the calculation of some intermediate terms or complicated iterative computing procedures. In general, I and J are functions of m_n, N_1, N_2, ψ, ϕ_n, h_a, h_b, r_b, F, x_1 and x_2 [7]; C_v and K_v are functions of Q_v, n_p and d; and C_m and K_m are functions F and λ [5]. Addendum modification coefficients x_1 and x_2 in the above formulation are treated as dependent variables in the sense that when a design (m_n, N_1, N_2, ψ) has been obtained, x_1 and x_2 are adjusted or determined to improve sliding conditions, pitting resistance and bending strength of the design [36]. The sum of addendum modifications is required to be zero so that positive and negative addendum modifications are made on pinion and gear respectively, while at the same time standard centre distance and pressure angle are maintained.

Boundary limits are terms which put limiting bounds on some independent or dependent variables such that the varying of these variables is restricted. ψ_{max}, $(m_F)_{min}$, $(N_1)_{min}$ and δ are boundary limits. Boundary limits are set by the design engineer or recommended values can be assumed. To keep the axial thrust low, single helical gears usually have a maximum helix angle ψ_{max}=20° [30]. In heavy engineering, a smaller maximum is preferred, ψ_{max}=15°. To achieve adequate helix overlap and smoother meshing continuity, face contact ratio is normally required to be greater than 1, e.g., $(m_F)_{min}$=1.1. In some applications, a minimum of 2 is required, $(m_F)_{min}$=2. $(N_1)_{min}$ must be at least the minimum number of teeth for avoidance of involute interference. For helical gears, $(N_1)_{min}$=17 would normally cover for involute interference. With some positive addendum modifications, the minimum could be as low as $(N_1)_{min}$=10 [28]. Because numbers of teeth of pinion and gear can only be whole numbers, it may not always be possible to achieve the exact gear ratio specified. Depending on the speed changeability of the prime mover, e.g. motor, and the acceptance of speed difference by the driven machine, gear ratio tolerance δ would normally be set between 1% and 5%. The introduction of gear ratio tolerance means that for a particular number of pinion teeth, the number of gear teeth may vary and still satisfy the gear ratio tolerance. It is for this reason that the number of gear teeth is included in the independent design variables.

Fixed parameters are terms whose values are given by the design engineer for a certain design configuration. These include the gear tooth system, pinion aspect ratio, gear materials, and gear accuracy grade. For the gear tooth system, the most commonly used for hob finished gears is ϕ_n=20, h_a=1, h_b=1.25 and r_b=0.25 or r_b=0.3. For grind finished gears, h_b and r_b are increased to h_b=1.35 or h_b=1.4 and r_b=0.35 or r_b=0.4 to cater for the extra depth needed for clearance of the grinding wheel. For the pinion aspect ratio λ, too small a value will result in a thin gear blank, especially if the gear ratio is large. For example, a gear ratio m_G=5 and pinion aspect ratio λ=0.2 will make the gear wheel diameter to face width ratio 25:1, which is not very good for taking the helical axial load. An unnecessarily small pinion aspect ratio λ will also make the centre distance larger. However, too large a value of λ will not necessarily bring down the centre distance, since load distribution across the face width will not be as good, i.e., the extra face may not take any load. The value of λ is usually taken between 0.4 and 1.0 for single helical gears [28], for good load distribution across the face width and a relatively small gear wheel diameter to face width ratio giving wheel stability. For double helical gears, λ can be as high as 1.75 because there is no axial load and a better alignment is achieved by a self-centring action of the apexes of the double helix on pinion and gear. For the gear design engineer, the fixed parameters are only fixed when the optimisation is being carried out. For example, one set of gear materials may be tried first and a corresponding optimum design found. Another set of gear materials may be tried next. The relative cost of each combination of materials and gear geometry is assessed and a decision can be made by the engineer.

Design specifications are normally supplied by the gear customer. The minimum information would include power to be transmitted P, pinion speed n_p, gear ratio m_G, and for what application the gears are used. Application factors C_a and K_a could be determined by the gear design engineer, in consultation with AGMA 6010 [6] or in discussion with the customer. Further information such as life requirement, reliability requirement and operating temperature, could be used to determine life factors C_L and K_L, reliability factors C_R and K_R, and temperature factors C_T and K_T. Otherwise, these factors would assume values of unity. In any case, these factors are fixed once the specifications are given.

The design constraints in the above formulation of the problem can be divided into two groups. Power capacities for pitting resistance and bending strength can be separated from the rest of the constraints which are of a geometrical nature. The evaluation of the capacity group is complex and time consuming, whereas that of the geometry group is relatively simple. A design that satisfies the geometry group is said to be geometrically feasible in this thesis. The evaluation of the capacity group should only be done for geometrically feasible designs during the optimisation process.

CHAPTER FOUR: OPTIMUM GEAR DESIGN ALGORITHM

4.1 *Basic Approach*

The optimum gear design problem as defined in Chapter Three can be solved by two different approaches. One is to use a general purpose algorithm in conjunction with the analysis model. The general procedure for solving an optimum design problem by this approach is as follows:

1. Find an initial design X_0 which can be an existing design or any combination of the values of design variables. Set the initial design as the current design;
2. Evaluate the current design by the analysis model, i.e., calculating the objective function value and all the constraint function values;
3. An optimum design algorithm is used to decide:
   a. what measures should be taken to bring the current design back to the feasible region if any of the constraints are violated;
   b. how to improve the current design, i.e., how the information given by the current design and sometimes previous designs, e.g., gradients of objective function and constraint functions, should be used to generate a new feasible design which has a lower objective function value;
   c. whether to terminate the optimisation process if no improvements is possible;
4. If a new design is generated by 3.a or 3.b, go back to step 2. If some terminating criteria are met in 3.c, end the optimisation process.

The procedure described above is often implemented in computer programs in a structure as depicted in Fig.4-1.

```
            The optimisation process
                         |
                 X=X_0, initial design
                         |
  /------------------->|
  |                      |
  |    analysis model: compute values of
  |        f(X), g_i(X) and h_i(X)
  |                      |
  |        optimum design algorithm:
  \---------compute new design X
                 or terminate
                         |
                 optimum design X*
```

Fig.4-1 General optimisation process

The development of mathematical techniques for solving optimum design problems have been on going for decades [39]. For the optimum design of engineering problems, the useful techniques are normally in the category of the so called nonlinear programming algorithms. The algorithms have been developed for continuous variable problems and they normally use the local information to make a move to achieve an improved design. Although use of the existing general purpose mathematical optimisation techniques can save effort on developing algorithms, difficulties have been reported when the general purpose algorithms were applied to the optimum design of gears. These include premature termination or stall of the algorithm, slow convergency speed, because of the highly nonlinear and complex procedures used in the gear rating analysis model. Adding to the difficulty of the problem, optimum gear design is a mixed discrete problem and the continuous variable algorithms are not really suitable for it. In any case, the general nonlinear algorithms can improve on a design but there is no guarantee that a true or global optimum design will result [49].

The alternative to the general purpose algorithm approach is a special purpose algorithm designed to be more efficient and effective for a particular optimum design problem. For example, exploiting the inherent relations in the gear analysis model to minimise the centre distance is typical of the special purpose algorithm approach.

The mathematical model of the optimum gear design problem as defined by equations (3-2-1) to (3-2-20) states clearly what the design variables are, what the objective function is, and what the constraint functions are. This formulation is often used in conjunction with a general purpose algorithm for the optimisation of the objective function. However, this initial formulation of the model is not very convenient to handle in the development of a special purpose algorithm.

Inspired by the K factor method [28] and the procedure used by Errichello [29] to solve for pinion pitch circle diameter, the capacity constraint equations (3-2-3) to (3-2-6) and (3-2-12) to (3-2-15) are singled out to be the most important constraints in determining the centre distance, although only pitting resistance capacities were used in [28] and [29]. Studying the simpler form equations (3-2-12a) to (3-2-15a) and considering the direct relation of equation (1-3-4) between centre distance C_r and pinion pitch circle diameter d, it can be said that if maximum and minimum values of geometry factors I and J are established, the capacity constraint equations can be used to set up lower and upper boundaries for the required centre distance and the minimum centre distance will lie between these boundaries. The factors C_v, K_v, C_m and K_m can be given initial values of unity and then updated by the closed form formulae (1-4-12), (1-4-13) and (1-4-14).

On the basis of the above statement, the original optimum gear design problem is to be solved at two levels. Firstly, a search interval for the centre distance is determined and a one dimensional search for the minimum centre distance is conducted within that interval. Secondly, for each trial value of the centre distance obtained during the one dimensional search, the maximum power combination of module, number of pinion teeth and number of gear teeth is found. The maximum permissible power capacity is then checked against the required power capacity in the one dimensional search to determine whether the trial centre distance can satisfy the capacity constraints.

Helix angle ψ in the above method is adjusted by equation (3-2-2) to make the centre distance C of the combination (m_n,N_1,N_2) equal to the trial centre distance. This in effect is an exchange of role between centre distance and helix angle as dependent variable and independent variable. Thus the four design variables (m_n,N_1,N_2,ψ) in the original formulation have now been replaced by four design variables $C,(m_n,N_1,N_2)$ at two levels. Centre distance C is now both the objective function and an independent variable. Helix angle ψ has been turned into a dependent variable.

To solve the optimum gear design problem by the above method, three sub-problems exist.

1. The determination of maximum and minimum geometry factors I and J to establish the search interval for centre distance;
2. The strategy for the one dimensional search of centre distance;
3. The strategy for determining the maximum power capacity combination of module, number of pinion teeth and number of gear teeth.

4.2 *Search Interval for Minimum Centre Distance*

To set up the search interval for the minimum centre distance, limiting values of geometry factors I and J must first be established. From equations (1-4-10) and (1-4-11) for geometry factors I and J, it can be seen that load sharing ratio m_N has a direct influence on the values of geometry factors.

4.2.1 *Limiting Values of Load Sharing Ratio*

Load sharing ratio m_N is defined by AGMA 908-B89 as:

$$m_N = \frac{F}{L_{min}} \qquad (4\text{-}2\text{-}1)$$

where F is the face width and L_{min} is the minimum total length of face contact lines. The physical meaning of m_N is the fraction of the total tangential load that one tooth is taking. For helical gears, L_{min} is calculated by AGMA as:

$$L_{min} = \frac{m_p F - n_a n_r p_x}{\cos\psi_b}, \qquad \text{if } n_a \le 1-n_r \qquad (4\text{-}2\text{-}2a)$$

$$L_{min} = \frac{m_p F - (1-n_a)(1-n_r)p_x}{\cos\psi_b}, \qquad \text{if } n_a > 1-n_r \qquad (4\text{-}2\text{-}2b)$$

where ψ_b is base helix angle, and n_r and n_a are fractional parts of transverse contact ratio m_p and face contact ratio m_F, respectively.

Substituting equation (4-2-2) for L_{min} in equation (4-2-1) and noticing equation (1-3-7), gives

$$m_N = \frac{\cos\psi_b}{m_p\left(1 - \dfrac{n_a n_r}{m_p m_F}\right)}, \qquad \text{if } n_a \le 1-n_r \qquad (4\text{-}2\text{-}3a)$$

$$m_N = \frac{\cos\psi_b}{m_p\left(1 - \dfrac{(1-n_a)(1-n_r)}{m_p m_F}\right)}, \qquad \text{if } n_a > 1-n_r \qquad (4\text{-}2\text{-}3b)$$

For any particular value of m_p, load sharing ratio m_N will have its maximum or minimum value when

$$\left(1 - \frac{n_a n_r}{m_p m_F}\right) \text{ or } \left(1 - \frac{(1-n_a)(1-n_r)}{m_p m_F}\right) \text{ is at a minimum or maximum.}$$

For convenience and simplicity, let

$$m_{ar} = \left(1 - \frac{n_a n_r}{m_p m_F}\right), \qquad \text{if } n_a \le 1-n_r \qquad (4\text{-}2\text{-}4a)$$

$$m_{ar} = \left(1 - \frac{(1-n_a)(1-n_r)}{m_p m_F}\right), \qquad \text{if } n_a > 1-n_r \qquad (4\text{-}2\text{-}4b)$$

Equation (4-2-3) can now be written as

$$m_N = \frac{\cos\psi_b}{m_p m_{ar}} \qquad (4\text{-}2\text{-}5)$$

Because m_{ar} reduces the effectiveness of transverse contact ratio m_p in equation (4-2-5), it may be called the *contact ratio derating factor*. Plotting equation (4-2-4) in Fig.4-2 shows the relationship of m_{ar} to m_p, with face contact ratio m_F as a parameter varying between 1 and 2. The curves have their minima when n_a=1-n_r or n_a+n_r=1. The locus of the minima is also shown in Fig.4-2.

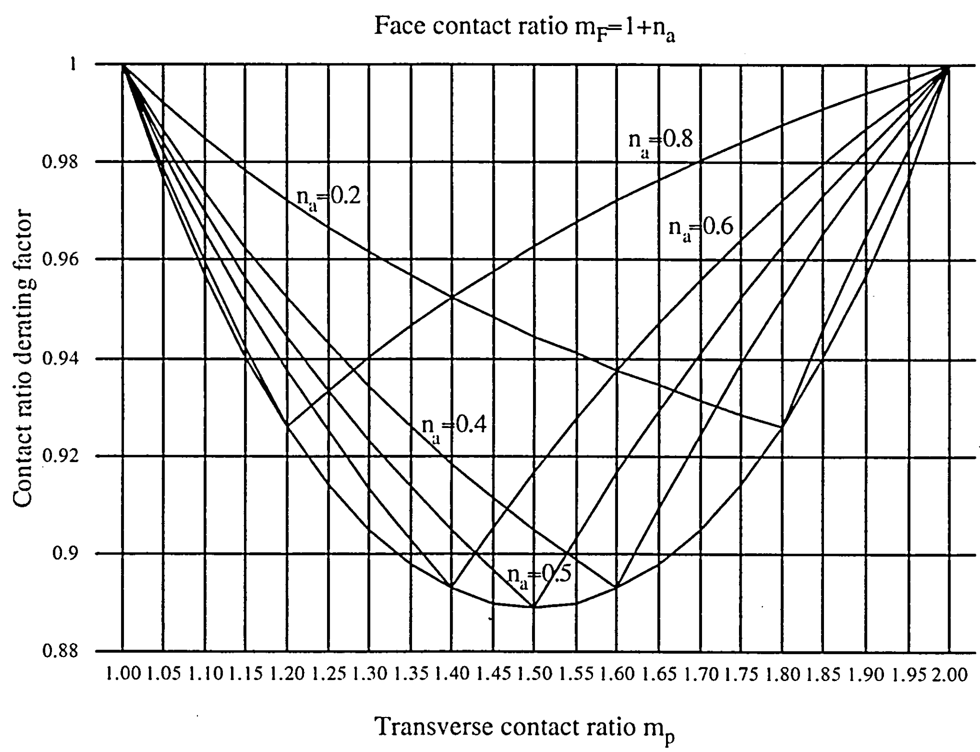


Fig.4-2 Contact ratio derating factor m_{ar}

Separating m_F into integer and fractional parts as m_F=I_a+n_a and given m_p, then m_{ar} can be considered as a function of n_a, while I_a is treated as a parameter. Differentiating equation (4-2-4a) with respect to n_a, gives

$$m_{ar}'(n_a) = - \frac{n_r m_p(I_a+n_a) - n_a n_r m_p}{[m_p(I_a+n_a)]^2}$$

simplifying,

$$m_{ar}'(n_a) = - \frac{n_r I_a}{m_p(I_a+n_a)^2}, \qquad \text{for } n_a \le 1-n_r \qquad (4\text{-}2\text{-}6a)$$

Since n_r, I_a, m_p and $(I_a+n_a)^2$ are all > 0, then $m_{ar}'(n_a)$ is < 0, which means m_{ar} monotonically decreases as n_a increases. For the valid range of $n_a \le 1-n_r$ for equations (4-2-4a) and (4-2-6a), m_{ar} will be at its minimum when n_a is at its maximum, i.e., when $n_a=1-n_r$. Conversely, m_{ar} will be at its maximum when n_a is at its minimum, i.e., when $n_a=0$. Similarly, differentiating equation (4-2-4b) with respect to n_a and simplifying, gives

$$m_{ar}'(n_a) = \frac{(1-n_r)(I_a+1)}{m_p(I_a+n_a)^2}, \qquad \text{for } n_a > 1-n_r \qquad (4\text{-}2\text{-}6b)$$

Since $(1-n_r)$, (I_a+1), m_p, and $(I_a+n_a)^2$ are all > 0, then $m_{ar}'(n_a)$ is > 0, which means m_{ar} monotonically increases as n_a increases. For the valid range of $n_a > 1-n_r$ for equations (4-2-4b) and (4-2-6b), m_{ar} will be at its minimum when n_a is at its minimum, i.e., when $n_a=1-n_r$. Conversely, m_{ar} will be at its maximum when n_a is at its maximum, i.e., when $n_a=1$.

Summarising the above derivations, m_{ar} is at a minimum when $n_a=1-n_r$ and m_{ar} is at a maximum when $n_a=0$ or $n_a=1$. Substituting $n_a=1-n_r$ into equations (4-2-4a) and (4-2-4b), gives

$$(m_{ar})_{min} = 1 - \frac{n_r(1-n_r)}{m_p(I_a+1-n_r)} \qquad (4\text{-}2\text{-}7)$$

Similarly, substituting $n_a=0$ into equation (4-2-4a) and $n_a=1$ into equation (4-2-4b), gives

$$(m_{ar})_{max} = 1 \qquad (4\text{-}2\text{-}8)$$

For standard tooth proportions or gears without excessive addendum modifications, $1 < m_p < 2$ and therefore, $n_r=m_p-1$. For helical gears with face contact ratio $m_F > 1$, I_a can take any integer value, but for the worst case of $I_a=1$, equation (4-2-7) becomes,

$$(m_{ar})_{min} = 1 - \frac{(m_p-1)(2-m_p)}{m_p(3-m_p)} \qquad (4\text{-}2\text{-}9)$$

Knowing $(m_{ar})_{min}$ and $(m_{ar})_{max}$, the maximum and minimum values of m_N are obtained by substituting equations (4-2-9) and (4-2-8) into equation (4-2-5) for m_{ar}. This gives

$$(m_N)_{max} = \frac{\cos\psi_b}{m_p\left(1 - \dfrac{(m_p-1)(2-m_p)}{m_p(3-m_p)}\right)} \qquad (4\text{-}2\text{-}10)$$

and

$$(m_N)_{min} = \frac{\cos\psi_b}{m_p} \qquad (4\text{-}2\text{-}11)$$

4.2.2 *Limiting Values of Geometry Factors*

When $\Sigma x=0$, as is specified in equation (3-2-16), the operating transverse pressure angle ϕ_r is the same as the standard transverse pressure angle ϕ. The limiting values or the minimum and maximum values of geometry factor I can be obtained by substituting ϕ for ϕ_r and equations (4-2-10) and (4-2-11) for m_N into equation (1-4-10), which is copied below,

$$I = \frac{\cos\phi_r \sin\phi_r}{2\ m_N} \frac{m_G}{m_G+1} \qquad (1\text{-}4\text{-}10)$$

resulting in,

$$I_{min} = \frac{\cos\phi\sin\phi}{2\cos\psi_b}\,\frac{m_G}{m_G+1}\,m_p\left(1 - \frac{(m_p-1)(2-m_p)}{m_p(3-m_p)}\right) \qquad (4\text{-}2\text{-}12)$$

and

$$I_{max} = \frac{\cos\phi\sin\phi}{2\cos\psi_b}\,\frac{m_G}{m_G+1}\,m_p \qquad (4\text{-}2\text{-}13)$$

The transverse contact ratio m_p in the above formulae can be calculated by

$$m_p = \frac{N_1}{\pi\cos\phi}\{[(\frac{1}{2} + \frac{h_a\cos\psi}{N_1})^2 - (\frac{\cos\phi}{2})^2]^{1/2} +$$

$$+ [(\frac{m_G}{2} + \frac{h_a\cos\psi}{N_1})^2 - (\frac{m_G\cos\phi}{2})^2]^{1/2} - \frac{1+m_G}{2}\sin\phi\} \qquad (4\text{-}2\text{-}14)$$

For the derivation of equation (4-2-14), see Appendix A. The transverse pressure angle ϕ and base helix angle ψ_b can be calculated by equations (1-3-10) and (1-3-11), respectively. Considering equations (4-2-14), (1-3-10) and (1-3-11), equations (4-2-12) and (4-2-13) expressed the minimum and maximum of geometry factor I as functions of gear ratio m_G, number of pinion teeth N_1, helix angle ψ, normal pressure angle ϕ_n, and addendum ratio h_a. Of these quantities, only N_1 and ψ are design variables, since ϕ_n and h_a are fixed design parameters once the gear tooth system has been chosen, and m_G is normally specified by the gear user or customer. For ϕ_n=20 and h_a=1, the relationships between the minimum and maximum geometry factor I and number of pinion teeth N_1, helix angle ψ, and gear ratio m_G are shown in Figs.4-3, 4-4 and 4-5.

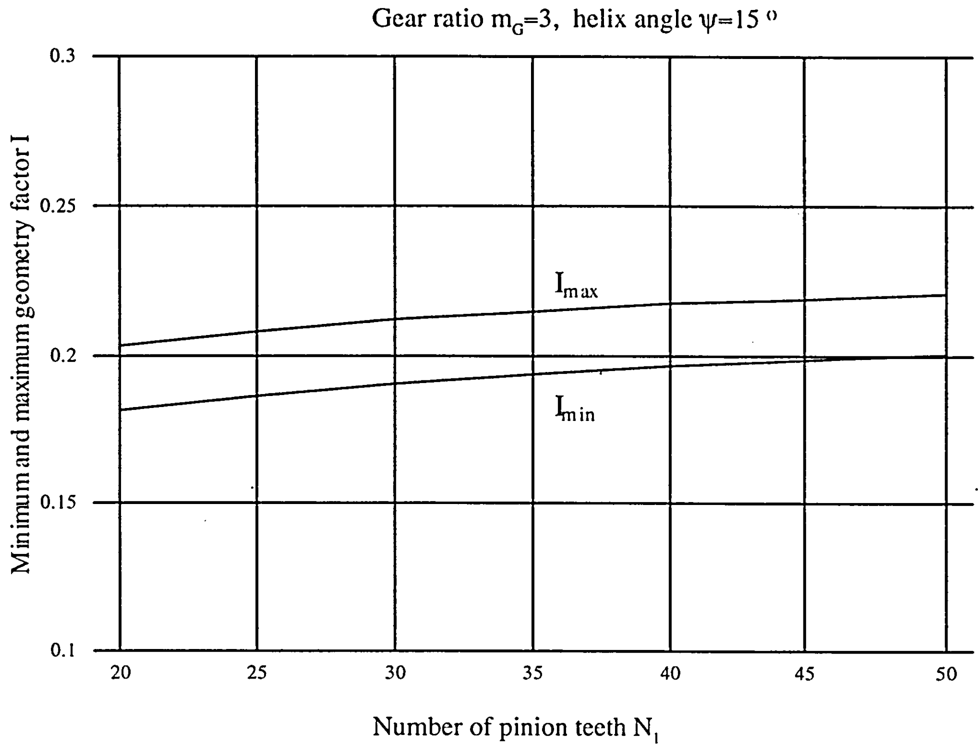


Fig.4-3 I_{min} and I_{max} as functions of N_1

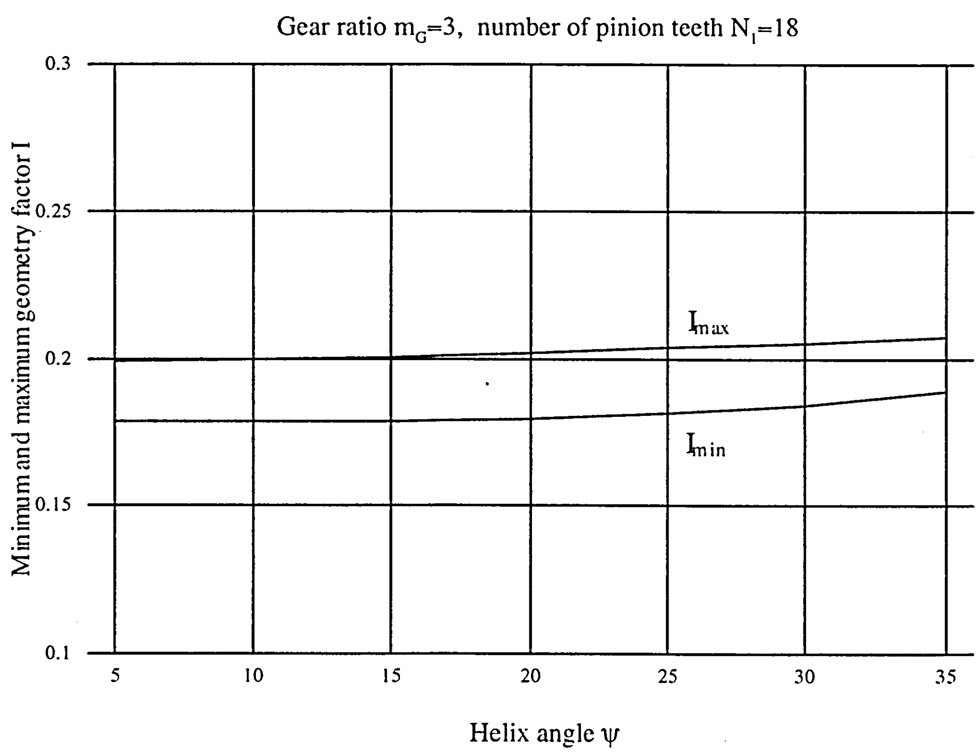


Fig.4-4 I_{min} and I_{max} as functions of ψ

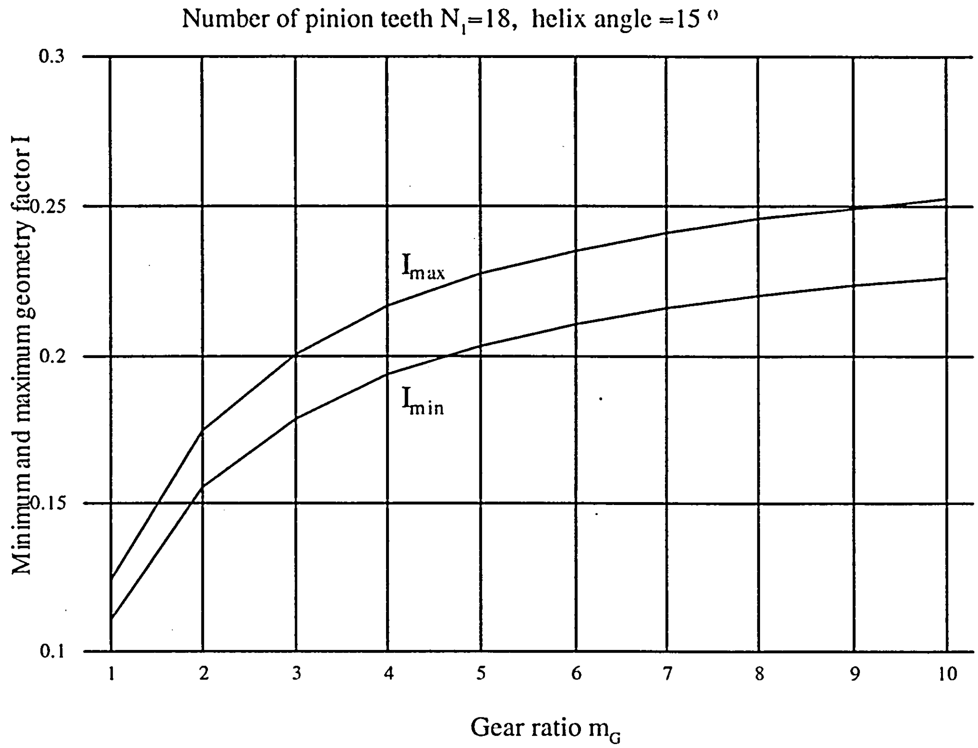


Fig.4-5 I_{min} and I_{max} as functions of m_G

It can be seen from these graphs that helix angle in the range shown does not have much effect on the minimum and maximum values of I. The number of pinion teeth has slightly larger effect but still cannot be considered significant. The major influence comes from the gear ratio m_G. It can be said that once the gear tooth system has been chosen and the gear ratio has been specified, the minimum and maximum geometry factor I can be roughly determined without knowing the number of pinion teeth N_1 and helix angle ψ. In any case, extreme values of N_1 and ψ can be used for the calculation of limiting values of geometry factor I. For instance, the minimum number of pinion teeth $(N_1)_{min}$ can be used to calculate I_{min} and a reasonably large number of pinion teeth, say 40 or 50 [28], can be used to calculate I_{max}.

For bending geometry factor J, as given in (1-4-11) and copied below,

$$J = \frac{Y}{K_f \; m_N} \qquad (1\text{-}4\text{-}11)$$

the values of Y and K_f are calculated by a complex iterative numerical procedure. Since the limiting values of m_N have already been derived, it is only necessary to consider Y/K_f to obtain the limiting values or the minimum and maximum values for geometry factor J. For helical gears with $m_F > 1$, geometry factor J is determined for load application at the tooth tip. This means that Y/K_f is a function of the geometry of one gear tooth only. Based on this understanding, the relationship between Y/K_f and gear tooth geometry can be revealed by curve fitting.

Since there are a discrete number of standard gear tooth systems, Y/K_f can be considered as a function of number of teeth N and helix angle ψ only, with standard normal pressure angle ϕ_n, addendum ratio h_a, dedendum ratio h_b, and fillet radius ratio r_b, being treated as fixed parameters. Fig.4-6 shows a plot of Y/K_f as a function of N for ψ=10. The gear tooth system used is ϕ_n=20°, h_a=1, h_b=1.25 and r_b=0.25. From the shape of the curve, it is quite obvious that Y/K_f is a function of the reciprocal of N.

It can be shown from the linear regression results, Table 4-1, that this function is linear:

$$\frac{Y}{K_f} = a + \frac{b}{N} \qquad (4\text{-}2\text{-}15)$$

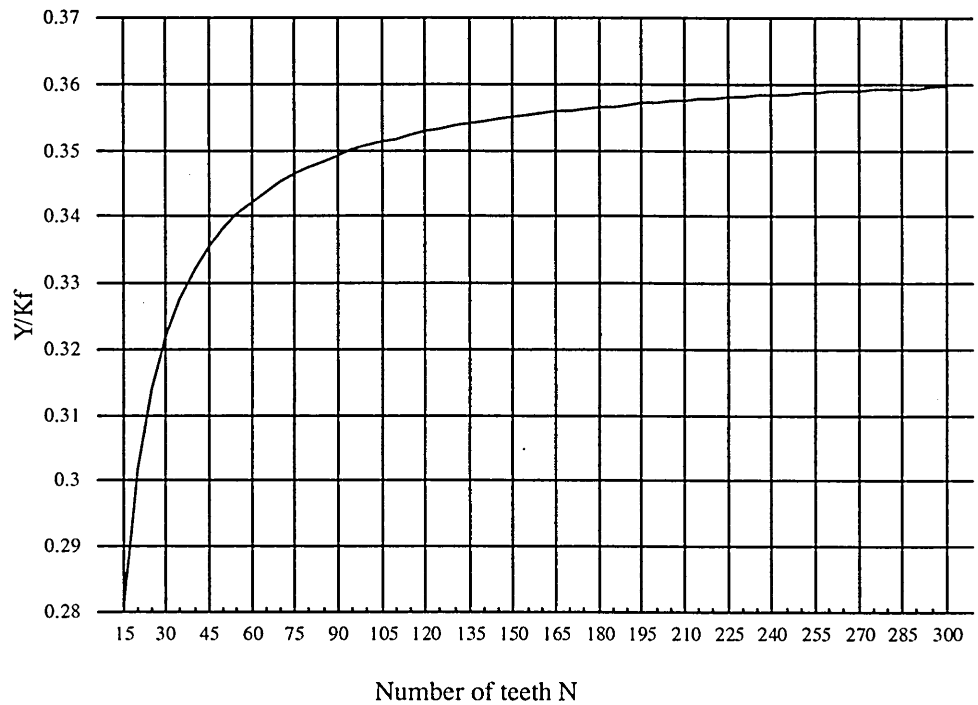


Fig.4-6 Y/K_f as a function of N

Table 4-1 Y/K_f as a linear function of 1/N

Helix Angle	5	10	15	20	25	30	35
constant a	0.347	0.3634	0.3697	0.3678	0.3584	0.3424	0.3203
coefficient b	-1.237	-1.241	-1.185	-1.079	-0.939	-0.778	-0.614
correl coef R squared	0.9996	0.9997	0.9997	0.9997	0.9997	0.9997	0.9996
std error of estimate	0.0003	0.0003	0.0002	0.0002	0.0002	0.0002	0.0001

As shown in Fig.4-7, plots of the constant a and coefficient b with respect to the helix angle ψ reveal trigonometric functions of the form:

$$a = a_1 + a_2\psi + a_3\sin(k\psi) + a_4\cos(k\psi) \qquad (4\text{-}2\text{-}16)$$

$$b = b_1 + b_2\psi + b_3\sin(k\psi) + b_4\cos(k\psi) \qquad (4\text{-}2\text{-}17)$$

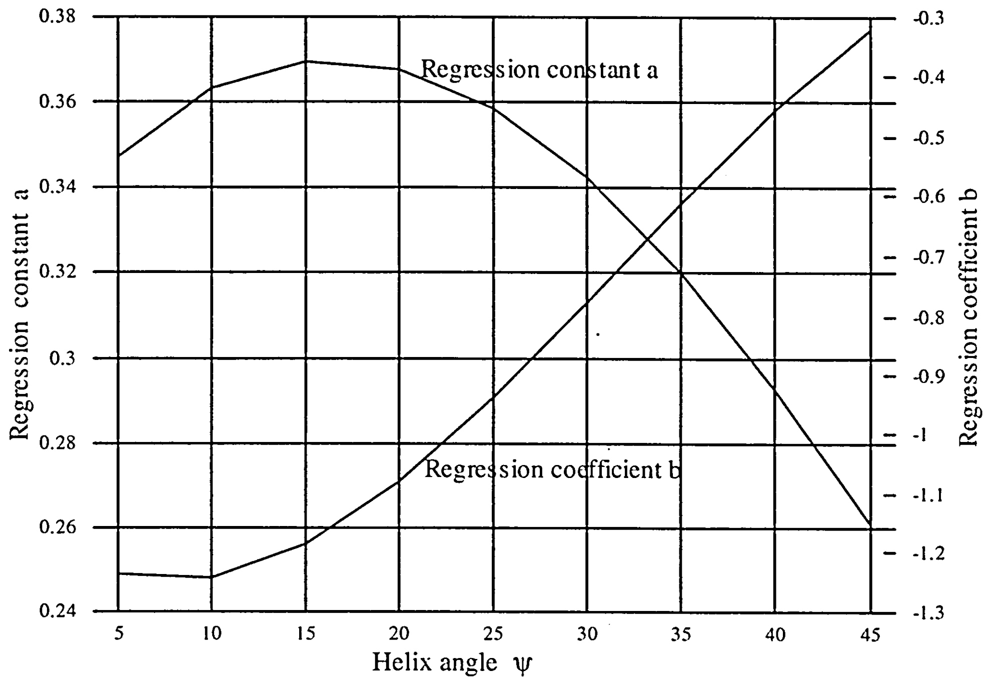


Fig.4-7 Regression constant a and coefficient b

Applying linear regression to the functions, good curve fittings can be found for the constant a and coefficient b.

Table 4-2 Regression constant a and coefficient b
Tooth system type 1: $\phi_n=20$, $h_a=1$, $h_b=1.25$, $r_b=0.25$
Tooth system type 2: $\phi_n=25$, $h_a=1$, $h_b=1.35$, $r_b=0.27$

Tooth		regression constant a				regression coefficient b			
System	k	a_1	a_2	a_3	a_4	b_1	b_2	b_3	b_4
Type 1	2.5	0.18167	0.000993	0.09731	0.14324	0.29927	-0.03157	0.25772	-1.46997
Type 2	2.5	0.24440	0.001402	0.15855	0.19416	0.57281	-0.05102	0.23135	-2.48014

Table 4-2 gives the constants and coefficients in equation (4-2-16) and (4-2-17) for two gear tooth systems. Thus the minimum and maximum values of geometry factor J for conventional helical gears can be obtained by substituting equation (4-2-15) for Y/K_f and equations (4-2-10) and (4-2-11) for m_N into equation (1-4-11) which results in the following equations:

$$J_{min} = \left(a + \frac{b}{N}\right) \frac{m_p}{\cos\psi_b} \left(1 - \frac{(m_p-1)(2-m_p)}{m_p(3-m_p)}\right) \qquad (4\text{-}2\text{-}18)$$

and

$$J_{max} = \left(a + \frac{b}{N}\right) \frac{m_p}{\cos\psi_b} \qquad (4\text{-}2\text{-}19)$$

where transverse contact ratio m_p is calculated from equation (4-2-14). Similar to the minimum and maximum of geometry factor I, equations (4-2-18) and (4-2-19) expressed the minimum and maximum of geometry factor J as functions of number of teeth N, helix angle ψ, gear ratio m_G, and gear tooth system ϕ_n, h_a, h_b, and r_b.

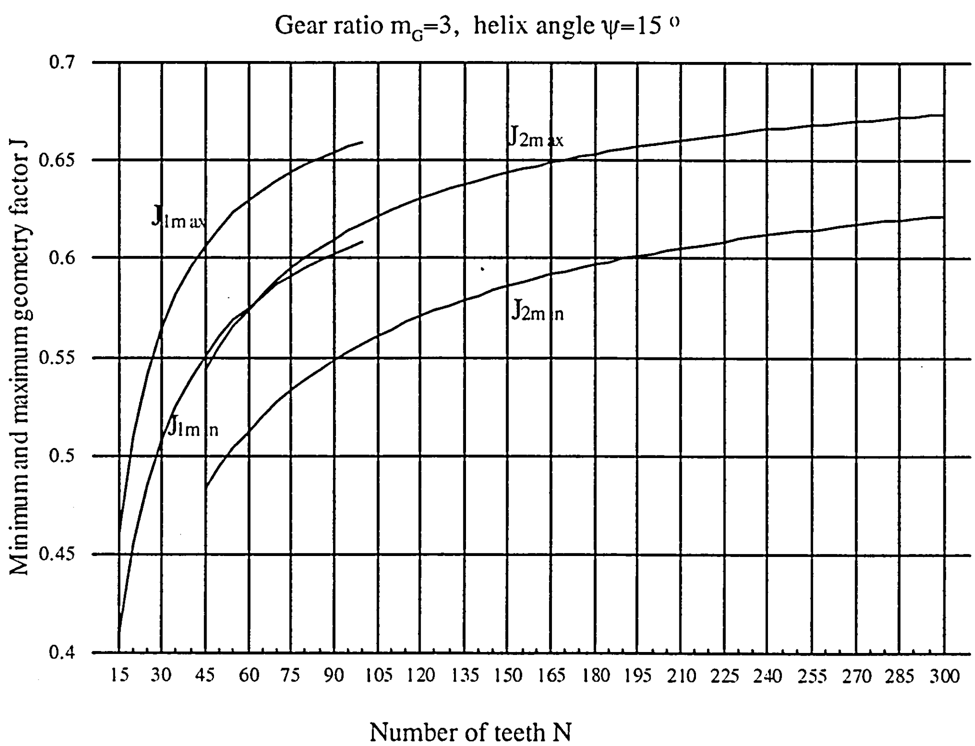


Fig.4-8 J_{min} and J_{max} as functions N

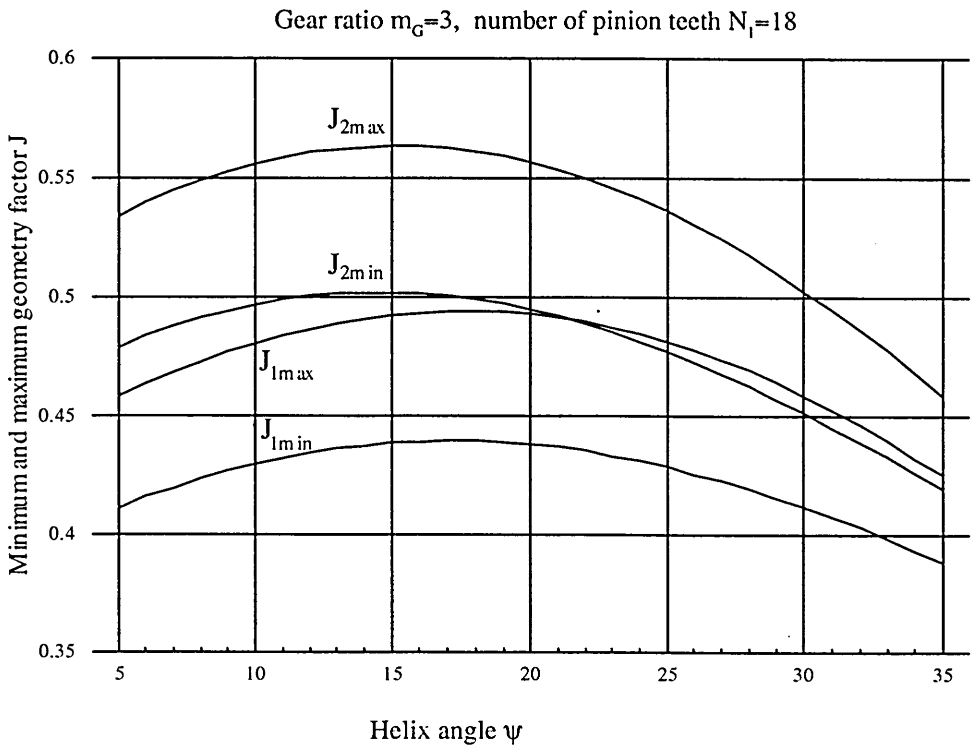


Fig.4-9 J_{min} and J_{max} as functions of ψ

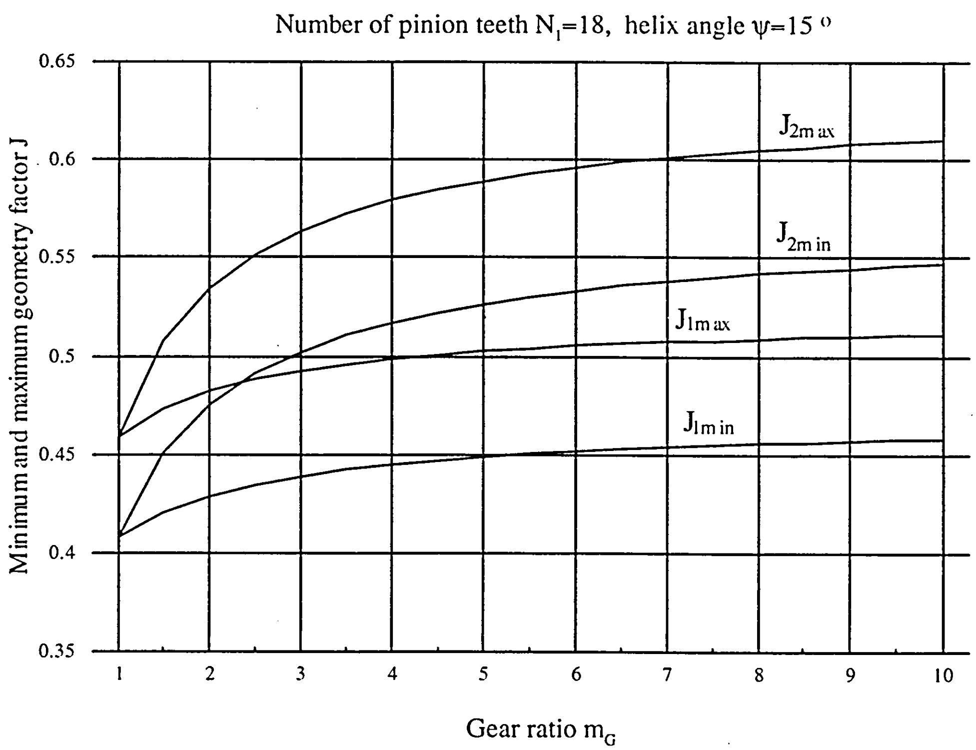


Fig.4-10 J_{min} and J_{max} as functions of m_G

Figs.4-8, 4-9, and 4-10 show the minimum and maximum values of geometry factor J as a function of N, ψ, and m_G. From Fig.4-8 it can be seen that the number of teeth N has a significant effect on the minimum and maximum values of J. For the effect of helix angle ψ on the minimum and maximum values of J as shown in Fig.4-9, J_{max} has its highest value around ψ=15°, and J_{min} has its lowest value when $\psi=\psi_{min}$ or $\psi=\psi_{max}$. The effect of gear ratio m_G is more significant for smaller gear ratios. Thus for the calculation of the minimum value of geometry factor J, minimum number of teeth should be used, which in the case of pinion is $(N_1)_{min}$ and in the case of gear is $m_G(N_1)_{min}$. For single helical gears, minimum helix angle should be used; whereas for double helical gears, maximum helix angle should be used. Conversely, for the calculation of the maximum value of geometry factor J, a reasonable larger number of teeth should be used, which in the case of pinion can be 50 and in the case of gear can be $50m_G$. Helix angle of 15° should be used for the calculation of J_{max}.

4.2.3 *Search Interval for Minimum Centre distance*

The minimum and maximum geometry factors I and J as given by equations (4-2-12), (4-2-13), (4-2-18) and (4-2-19) are expressed as functions of helix angle ψ and numbers of teeth N_1 and N_2 of pinion and gear, with gear ratio and gear tooth system being considered as parameters. These equations cannot be used directly because N_1, N_2 and ψ are design variables and they are not known until the optimum design has been done. To use these equations to establish suitable lower and upper boundaries for the search interval for the minimum centre distance, extreme values of possible numbers of teeth and helix angle are required, as suggested in the previous subsection.

Combining equations (3-2-3) and (3-2-12a), the pitting resistance constraints of the pinion, gives,

$$P \le C_{w1}\, n_p \frac{C_v}{C_m} I\, F\, d^2 \tag{4-2-20}$$

Pinion aspect ratio constraint (3-2-19) can be rewritten as

$$F = \lambda d \tag{4-2-21}$$

Noticing $C=C_r$ when addendum modification constraint (3-2-16) is satisfied, equation (1-3-4) can be rewritten as

$$d = \frac{2C}{m_G + 1} \tag{4-2-22}$$

Substituting equation (4-2-21) for F and (4-2-22) for d into equation (4-2-20) and rearranging, gives

$$C \ge \frac{1+m_G}{2}\left(\frac{P}{n_p C_{w1}}\,\frac{C_m}{C_v}\,\frac{1}{\lambda I}\right)^{1/3} \tag{4-2-23}$$

Equation (4-2-23) gives the centre distance required to transmit the specified power P, at speed n_p and gear ratio m_G, based on the amalgamated factor C_{w1} for pinion pitting resistance. The minimum centre distance could be obtained by using the equal sign in equation (4-2-23):

$$C_{min} = \frac{1+m_G}{2}\left(\frac{P}{n_p C_{w1}}\,\frac{C_m}{C_v}\,\frac{1}{\lambda I}\right)^{1/3} \tag{4-2-24}$$

The problem is C_m, C_v and I are unknowns and are functions of gear geometry. This difficulty is dealt with as follows.

When I takes its upper limiting value, i.e., $I=I_{max}$, equation (4-2-24) gives the lower boundary for the minimum centre distance,

$$C^{L}_{min}=\frac{1+m_G}{2}\left(\frac{P}{n_p C_{w1}}\,\frac{C_m}{C_v}\,\frac{1}{\lambda I_{max}}\right)^{1/3} \qquad (4\text{-}2\text{-}25a)$$

In other words, the minimum centre distance C_{min} must be larger than C^{L}_{min}. Similarly, when I takes its lower limiting value, i.e., $I=I_{min}$, equation (4-2-24) gives the upper boundary for the minimum centre distance,

$$C^{U}_{min}=\frac{1+m_G}{2}\left(\frac{P}{n_p C_{w1}}\,\frac{C_m}{C_v}\,\frac{1}{\lambda I_{min}}\right)^{1/3} \qquad (4\text{-}2\text{-}26a)$$

In other words, the minimum centre distance C_{min} can only be smaller than C^{U}_{min}.

Similar equations can be derived for the gear,

$$C^{L}_{min}=\frac{1+m_G}{2}\left(\frac{P}{n_p C_{w2}}\,\frac{C_m}{C_v}\,\frac{1}{\lambda I_{max}}\right)^{1/3} \qquad (4\text{-}2\text{-}25b)$$

and

$$C^{U}_{min}=\frac{1+m_G}{2}\left(\frac{P}{n_p C_{w2}}\,\frac{C_m}{C_v}\,\frac{1}{\lambda I_{min}}\right)^{1/3} \qquad (4\text{-}2\text{-}26b)$$

which are based on equations (3-2-4) and (3-2-13a), the pitting resistance constraints of the gear.

When equations (4-2-25) and (4-2-26) are used to establish the lower and upper boundaries for the minimum centre distance, dynamic factor C_v and load distribution factor C_m are assigned an initial value of 1. After calculation of the lower and upper boundaries C^{L}_{min} and C^{U}_{min}, new values of C_v and

C_m are obtained using AGMA closed form formulae (1-4-13) and (1-4-14). C^L_{min} and C^U_{min} are then recalculated. In 3 or 4 iterations, C_v and C_m will converge to an accuracy better than 0.01 which is generally acceptable.

Equations (4-2-25) and (4-2-26) are based on the pitting resistance constraints. When the bending strength is more critical, equations (3-2-5),(3-2-6),(3-2-14a) and (3-2-15a), the bending strength constraints of the pinion and gear, can be similarly transformed. From (3-2-5) and (3-2-14a), the following can be obtained,

$$P \le K_{w1}\, n_p \frac{K_v}{K_m} J_1\, F \frac{d}{P_d} \tag{4-2-27}$$

Combining equations (3-2-17) and (3-2-18), gives

$$P_d = \frac{N_1}{d} \tag{4-2-28}$$

Substituting equation (4-2-28) for P_d, (4-2-21) for F and (4-2-22) for d into equation (4-2-27) and rearranging, gives

$$C \ge \frac{1+m_G}{2} \left(\frac{P}{n_p K_{w1}} \frac{K_m}{K_v} \frac{N_1}{\lambda J_1} \right)^{1/3} \tag{4-2-29}$$

The minimum centre distance based on pinion bending strength then is,

$$C_{min} = \frac{1+m_G}{2} \left(\frac{P}{n_p K_{w1}} \frac{K_m}{K_v} \frac{N_1}{\lambda J_1} \right)^{1/3} \tag{4-2-30}$$

In this case, a tooth size as large as possible should be used to obtained a search interval as small as possible. In other words, the minimum number of teeth $(N_1)_{min}$ should be used, as follows,

$$C_{min} = \frac{1+m_G}{2}\left(\frac{P}{n_p K_{w1}}\,\frac{K_m}{K_v}\,\frac{(N_1)_{min}}{\lambda J_1}\right)^{1/3} \qquad (4\text{-}2\text{-}31)$$

Substituting the limiting values of geometry factor J_{1max} and J_{1min}, the lower and upper boundaries for the minimum centre distance are,

$$C^L_{min} = \frac{1+m_G}{2}\left(\frac{P}{n_p K_{w1}}\,\frac{K_m}{K_v}\,\frac{(N_1)_{min}}{\lambda J_{1max}}\right)^{1/3} \qquad (4\text{-}2\text{-}32a)$$

and

$$C^U_{min} = \frac{1+m_G}{2}\left(\frac{P}{n_p K_{w1}}\,\frac{K_m}{K_v}\,\frac{(N_1)_{min}}{\lambda J_{1min}}\right)^{1/3} \qquad (4\text{-}2\text{-}33a)$$

Similar equations based on the bending strength of the gear are,

$$C^L_{min} = \frac{1+m_G}{2}\left(\frac{P}{n_p K_{w2}}\,\frac{K_m}{K_v}\,\frac{(N_1)_{min}}{\lambda J_{2max}}\right)^{1/3} \qquad (4\text{-}2\text{-}32b)$$

and

$$C^U_{min} = \frac{1+m_G}{2}\left(\frac{P}{n_p K_{w2}}\,\frac{K_m}{K_v}\,\frac{(N_1)_{min}}{\lambda J_{2min}}\right)^{1/3} \qquad (4\text{-}2\text{-}33b)$$

The factors K_v and K_m in equations (4-2-32) and (4-2-33) are obtained by the same iterative method as for C_v and C_m.

Four values are obtained each for the lower and upper boundaries for the minimum centre distance. The largest of each four values should be used for either of the boundary. In the case of the lower boundary, it is based on the maximum possible values of geometry factors, the actual geometry factors can only be smaller and hence the centre distance can only be larger. The largest value of the lower boundary suggests that there is a need for the centre distance to be at least as large as this value, whether this need comes from pinion pitting resistance, pinion bending strength, gear pitting resistance, or gear bending strength. When the lower boundary takes this largest value, the other lower values will automatically be covered. In the case of the upper boundary, it is based on the minimum possible values of geometry factors. The largest value of the upper boundary suggests that the centre distance may be required to be as large as this value and certainly there is no need to be larger than this value. Thus using the largest value of the upper boundary can cover for all possible requirements.

4.3 *One Dimensional Search for Minimum Centre Distance*

Once the lower and upper boundaries for the search interval of the minimum centre distance have been established, a macro or general monotonic relation between the centre distance and its power capacity can be exploited to minimise the centre distance. Fig.4-11 shows this relation for a particular example. The power capacity shown for any centre distance is the maximum power capacity of all geometrically feasible combinations of module and numbers of teeth of pinion and gear at that centre distance. This macro monotonic relation can be explained by the AGMA power capacity rating formulae.

Substituting equations (4-2-21) for F and (4-2-22) for d into power rating equation (3-2-12a) and rearranging, gives

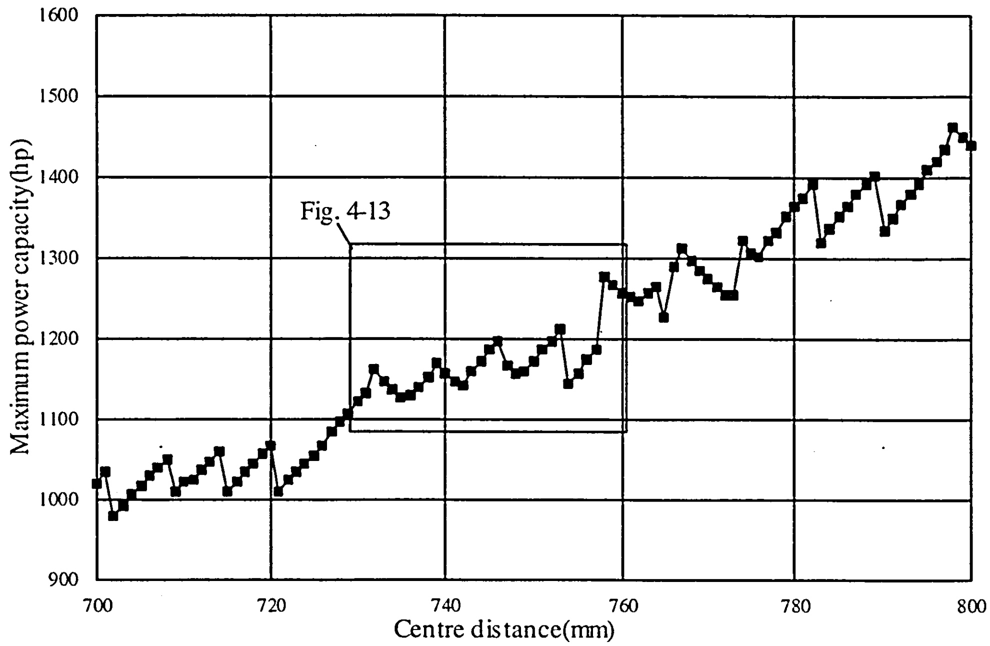


Fig.4-11 Centre distance and its maximum power capacity

$$P_{ac1} = C_{w1}\, n_p \frac{C_v}{C_m} I\lambda \frac{8C^3}{(m_G+1)^3} \qquad (4\text{-}3\text{-}1)$$

The equation shows power capacity rating is a cubic function of centre distance C. Whilst the factors C_v and C_m are variable, their rates of change are of a much lower order with respect to the centre distance C. For instance, from Fig.1-12, even for the highest rate of change of C_v, say when pitch line velocity v_t is doubled from 200 ft/min to 400 ft/min, which means the centre distance is doubled, C_v for AGMA quality grade Q_v=5 is only changed from 0.78 to 0.71, which is a reduction of less than 10%. As for the geometry factor I, its range of change is very limited when the gear ratio is given. See Fig.4-3 and Fig.4-4. Considering the lower rates of change of C_v and C_m and the limited range of change of I, equation (4-3-1) will give higher power capacity rating for pinion pitting resistance when the increment in centre distance is large. A similar conclusion can be reached for power capacity rating for gear pitting resistance. When bending strength is more critical, equation (3-2-14a) can be similarly transformed into,

$$P_{at1} = K_{w1}\, n_p \frac{K_v}{K_m} \frac{J_1}{N_1} \lambda \frac{8C^3}{(m_G+1)^3} \qquad (4\text{-}3\text{-}2)$$

which demonstrates that power capacity rating for pinion bending strength is also a cubic function of centre distance C. The factors K_v and K_m are the same as C_v and C_m. They have a lower rate of change than the centre distance. To find the maximum power of a particular centre distance, when bending strength is more critical, the minimum number of pinion teeth $(N_1)_{min}$ is used. This eliminates N_1 in (4-3-2) as a variable. At the same time, when gear ratio has been given and number of pinion teeth has been decided, geometry factor J_1 has a very limited range of change. See Fig.4-9. This leads to a similar conclusion that equation (4-3-2) will give higher power capacity rating for pinion bending strength when the increment in centre distance is large. A similar conclusion can be reached for power capacity rating for gear bending strength.

The above relationship between the centre distance and its power capacity rating is called macro monotonicity because the power capacity rating increases as the centre distance increases, when the magnitude of change is relatively large. When the magnitude of change is small, it can be seen in

Fig.4-11 that there exists local or micro non-monotonicity. Micro non-monotonicity will be discussed in Section 4.5.

Considering the macro monotonicity, it can be said that if a centre distance has a power capacity rating greater than the required power, all larger centre distances will satisfy the power requirement. This means that a centre distance satisfying the power capacity constraints can be used as a new upper boundary of the search interval. Similarly, a centre distance not satisfying the power capacity constraints can be used as a new lower boundary of the search interval, since any smaller centre distance will have smaller power capacity rating. Thus the strategy for the one dimensional search for the minimum centre distance is as follows. Using the lower and upper boundaries established, a middle value can be calculated for the centre distance. If this centre distance satisfies the power capacity constraints, it is used as the new upper boundary. Otherwise, it is used as the new lower boundary. This process is repeated until the difference between the boundaries is less than a pre-set tolerance. A local search is needed after the one dimensional search to eliminate the effect of micro non-monotonicity which will be discussed later. To summarise, the one dimensional search for the minimum centre distance based on the macro monotonicity can be presented in the following steps:

1. Establish the lower and upper boundaries of the minimum centre distance for the power, speed and gear ratio requirement, using equations (4-2-25), (4-2-26), (4-2-32), and (4-2-33);

2. Calculate the mean value of the lower and upper boundaries of the minimum centre distance;
3. Find the maximum power capacity rating for the mean centre distance;
4. If the power capacity rating is larger than the required power capacity, replace the upper boundary by the mean value; otherwise, replace the lower boundary by the mean value;
5. Calculate the difference between the lower and upper boundaries;
6. If the difference is greater than a pre-set tolerance, go to step 2; otherwise, terminate the iteration;
7. The upper boundary is the minimum centre distance based on macro monotonicity.

The strategy is very efficient for two reasons. First, for both I and J, the differences between the maximum and minimum values are not large, resulting in a small search interval to start with. Second, each trial value of centre distance at the middle point can discard half of the search interval, thus reducing the search interval very rapidly. But before the strategy can be implemented, a method for solving the problem posed by step 3 in the above procedure has to be established.

4.4 *Maximum Power Capacity of Fixed Centre Distance*

For every trial value of centre distance in the above strategy, the combination of module and numbers of teeth of pinion and gear (m_n,N_1,N_2) which gives the maximum power capacity at that centre distance has to be found. This combination must satisfy all the geometry constraints. The geometry constraints are satisfied by transforming them into boundary constraints on the design variables of module and numbers of teeth of pinion and gear. This is a more efficient way of treating the constraints since they can be automatically satisfied when the design variables take values within the boundaries. The maximum power combination is found by enumerating all possible combinations. The enumeration takes place in the order of module m_n, number of teeth of pinion N_1, and number of teeth of gear N_2. For typical values of geometry constraints, the transformed boundaries on module and numbers of teeth of pinion and gear are fairly tight, resulting in a small number of combinations.

The upper boundary on module is determined by two geometry constraints. They are the minimum face contact ratio $(m_F)_{min}$ and the minimum number of pinion teeth $(N_1)_{min}$. The minimum face contact ratio constraint, equation (3-2-8) and equation (3-2-20), can be rewritten as

$$(m_F)_{min} \le \frac{F\sin\psi}{\pi m_n} \tag{4-4-1}$$

which using equations (4-2-21) and (4-2-22), can be transformed into an upper boundary on module,

$$m^U_n = \frac{2\lambda C}{1 + mG} \frac{\sin\psi_{max}}{\pi (m_F)_{min}} \tag{4-4-2}$$

where C is a trial value for the minimum centre distance. When centre distance is at a trial value, equation (3-2-2) becomes a constraint that has to be satisfied by the module,

$$m_n = \frac{2C\cos\psi}{N_1 + N_2} \tag{4-4-3}$$

Using the minimum number of pinion teeth $(N_1)_{min}$ and assuming that $\cos\psi$ has a maximum value of 1, equation (4-4-3) can be transformed into an upper boundary on module,

$$m^U{}_n = \frac{2C}{(N_1)_{min}+(N_2)_{min}} \tag{4-4-4}$$

where $(N_2)_{min}$ is the minimum number of gear teeth when number of pinion teeth is $(N_1)_{min}$, which is derived from gear ratio tolerance constraint (3-2-11),

$$(N_2)_{min} = (1-\delta)m_G(N_1)_{min} \tag{4-4-5}$$

The value given by equation (4-4-5) is rounded up to the nearest whole number. The upper boundary on the module is the smaller of the values given by equations (4-4-2) and (4-4-4).

The lower boundary on module is imposed by bending strength. The bending strength constraints (3-2-5) and (3-2-14a) have previously been combined into equation (4-2-27). Noticing equation (3-2-18) can be rewritten as

$$P_d = \frac{\cos\psi}{m_n} \tag{4-4-6}$$

then substituting equations (4-4-6) for P_d, (4-2-21) for F, and (4-2-22) for d into equation (4-2-27) and rearranging, gives

$$m_n \geq \frac{P}{K_w n_p} \frac{K_m}{K_v} \frac{\cos\psi}{J_1} \frac{(1+m_G)^2}{4\lambda C^2} \qquad (4\text{-}4\text{-}7)$$

The minimum value or the lower boundary of module $m^L{}_n$ is reached when $J_1=J_{1max}$ and $\psi=\psi_{max}$, as follows,

$$m^L{}_n = \frac{P}{K_w n_p} \frac{K_m}{K_v} \frac{\cos\psi_{max}}{J_{max}} \frac{(1+m_G)^2}{4\lambda C^2} \qquad (4\text{-}4\text{-}8)$$

K_v and K_m are calculated for the pitch line velocity and face width related to the trial value of the centre distance. Equations (4-4-2), (4-4-4) and (4-4-8) indicate that as the trial value of the centre distance C becomes smaller in the minimisation process, the upper and lower boundaries on the module move closer to each other, which means fewer modules are to be tried for each trial centre distance and an acceleration of convergence.

For each module within the boundaries established by the above equations, lower and upper boundaries on number of pinion teeth N_1 and number of gear teeth N_2 can be found by transforming the geometry constraints. The gear ratio tolerance can be transformed into lower and upper boundaries $N^L{}_2$ and $N^U{}_2$ on number of gear teeth N_2 as follows,

$$N^L{}_2 = (1-\delta)m_G N_1 \qquad (4\text{-}4\text{-}9)$$

and

$$N^U{}_2 = (1+\delta)m_G N_1 \qquad (4\text{-}4\text{-}10)$$

The values given by equations (4-4-9) and (4-4-10) are rounded up and down respectively.

When the centre distance and module have each taken trial values, equation (3-2-2) becomes a constraint on the total number of teeth (N1+N2), which can be written as

$$(N_1+N_2) = \frac{2C\cos\psi}{m_n} \qquad (4\text{-}4\text{-}11)$$

then the lower and upper boundary on (N_1+N_2) are

$$(N_1+N_2)^L = \frac{2C\cos\psi_{max}}{m_n} \qquad (4\text{-}4\text{-}12)$$

and

$$(N_1+N_2)^U = \frac{2C\cos\psi_{min}}{m_n} \qquad (4\text{-}4\text{-}13)$$

where ψ_{min} is the minimum helix angle to satisfy the minimum face contact ratio constraints (3-2-8) and (3-2-20), or (4-4-1), when centre distance and module are given, and equations (4-2-21) and (4-2-22) are considered, hence

$$\psi_{min} = \arcsin\left(\frac{\pi m_n (m_F)_{min}(1+m_G)}{2\lambda C}\right) \qquad (4\text{-}4\text{-}14)$$

Again, the values given by equations (4-4-12) and (4-4-13) are rounded up and down, respectively.

Obviously, the lower boundary $N^L{}_1$ on number of pinion teeth N_1 can be found when (N_1+N_2) is at the lower boundary given by equation (4-4-12) and N_2 is at the upper boundary given by equation (4-4-10), i.e.,

$$N^L{}_1 + (1+\delta)m_G N^L{}_1 = (N_1+N_2)^L \qquad (4\text{-}4\text{-}15)$$

The above equation can be solved for $N^L{}_1$, resulting in,

$$N^L{}_1 = \frac{(N_1+N_2)^L}{1+(1+\delta)m_G} \qquad (4\text{-}4\text{-}16)$$

The value obtained is rounded up. Similarly, the upper boundary N^U_1 on number of pinion teeth N_1 is found to be,

$$N^U_1 = \frac{(N_1+N_2)^U}{1+(1-\delta)m_G} \qquad (4\text{-}4\text{-}17)$$

The value obtained is rounded down.

Having established the lower and upper boundaries on module and numbers of teeth of pinion and gear, the strategy for finding the combination (m_n,N_1,N_2) which gives the maximum power capacity at a given centre distance can now be summarised as follows.

First, the lower and upper boundaries m^L_n and m^U_n on module m_n are established, using equations (4-4-8), (4-4-2) and (4-4-4). The boundaries are dependent on the centre distance C.

Second, for each trial value of module m_n within the boundaries m^L_n and m^U_n, the lower and upper boundaries N^L_1 and N^U_1 on number of pinion teeth N_1 are established, using equations (4-4-16) and (4-4-17). The boundaries are dependent on the centre distance C and module m_n.

Third, for each trial value of number of pinion teeth N_1 within the boundaries N^L_1 and N^U_1, the lower and upper boundaries N^L_2 and N^U_2 on number of gear teeth N_2 are established, using equations (4-4-9) and (4-4-10). The boundaries are dependent on the centre distance C, module m_n, and number of pinion teeth N_1.

When a combination of module, number of pinion teeth and number of gear teeth (m_n,N_1,N_2) is synthesized using the above sequence, the combination will automatically satisfy the geometry constraints. The power capacities of all possible combinations within the boundaries are calculated and the combination with the maximum power capacity is found by simple comparison. For the purpose of reducing the search interval of centre distance more efficiently during the minimisation process, it is not necessary to try all possible combinations, because as soon as a combination having a power capacity larger than the power requirement is found, the trial value of the centre distance can be used as the new upper boundary of the minimum centre distance.

4.5 *Local Minimisation of Centre Distance*

It is noted in Section 4.3 that micro non-monotonicity exists when the magnitude of change in centre distance is small, as shown in Fig.4-11. The micro non-monotonicity is so called because the maximum power capacity rating may not necessary increase if the centre distance is increased only by a small amount. The minimum centre distance obtained by the one dimensional search is based on the macro monotonicity. When the one dimensional search terminates, the centre distance may or may not be at the true minimum or the global minimum, although it is close to the global minimum since micro non-monotonicity only exists in a local sense. Thus to find the global minimum after the one dimensional search terminates, a local search is needed to eliminate the effect of micro non-monotonicity.

There are two micro non-monotonic behaviours shown in Fig.4-11. One is the step change and the other is the valley in the maximum power capacity ratings. These two non-monotonic behaviours are examined below and an algorithm is devised accordingly for the local search.

For a combination (m_n, N_1, N_2), the centre distance C is related to the helix angle ψ by equation (3-2-2),

$$C = \frac{(N_1+N_2)m_n}{2\cos\psi} \qquad (3\text{-}2\text{-}2)$$

Since the variation of helix angle ψ is limited, the variation of centre distance C for a combination (m_n, N_1, N_2) is also limited. In other words, there is a centre distance range associated with a combination (m_n, N_1, N_2). From equation (3-2-2), it can be seen that the lower limit of the centre distance range is determined by the minimum helix angle ψ_{min}. The minimum helix angle ψ_{min} is in turn determined by the minimum face contact ratio constraints (3-2-8) and (3-2-20), or (4-4-1), when number of pinion teeth N_1 is known, and equations (3-2-17) and (3-2-19) are considered,

$$\psi_{min} = \arctan\left(\frac{\pi (m_F)_{min}}{\lambda N_1}\right) \qquad (4\text{-}5\text{-}1)$$

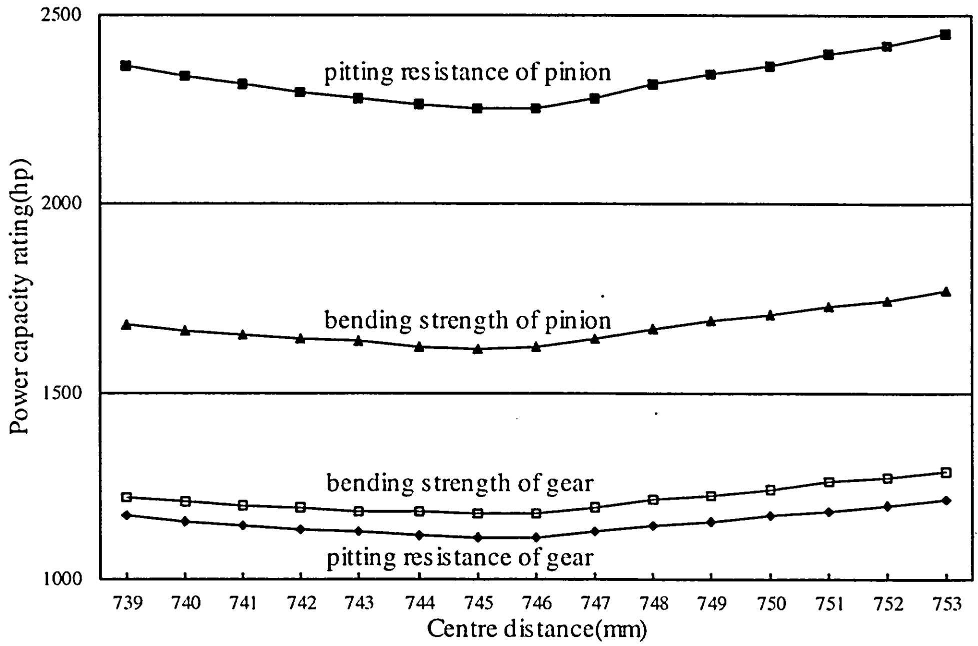
Thus the lower and upper limits of the centre distance range of a combination (m_n, N_1, N_2) are,

$$C^L{}_{range} = \frac{(N_1+N_2)m_n}{2\cos\psi_{min}} \qquad (4\text{-}5\text{-}2)$$

and

$$C^U{}_{range} = \frac{(N_1+N_2)m_n}{2\cos\psi_{max}} \qquad (4\text{-}5\text{-}3)$$

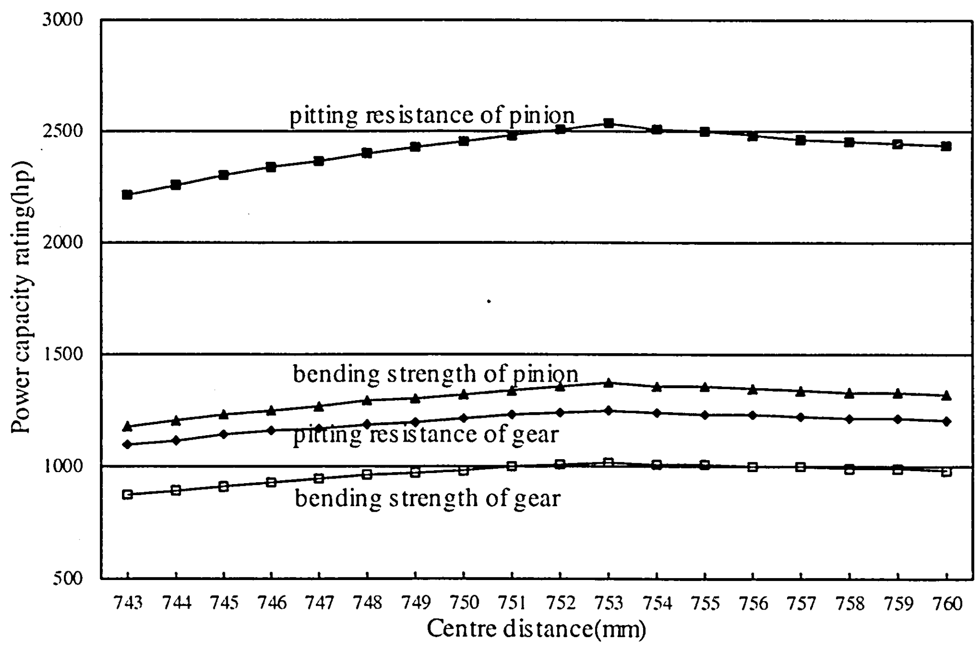
$(m_n, N_1, N_2) = (14, 20, 84)$



(a)

Pinion case hardened, gear through hardened

$(m_n, N_1, N_2) = (10, 28, 119)$



(b)

Fig.4-12 Power capacity ratings versus centre distance

Typical curves showing the relationship between the power capacity ratings and the centre distance of individual combinations are shown in Fig.4-12, where only the lowest of the four curves represents the permissible power capacity. Note that the numerical values of centre distance C and normal module m_n are and will be in mm. Table 4-3 shows the data used to draw Fig.4-12 together with the values of helix angle ψ, face contact ratio m_F and transverse contact ratio m_p. Note that for Fig.4-12(a), the minimum power capacity ratings occur at C=745 and C=746 when $m_F+m_p=1.98$ and $m_F+m_p=2.01$. For Fig.4-12(b), the maximum power capacity ratings occur at C=753 when $m_F=2.01$. Thus the maximum power capacity rating of an individual combination is achieved either at the lower or upper limits of the centre distance range or at a point where face contact ratio m_F is a whole number. Referring back to equations (4-3-1) and (4-3-2) and the discussions about C_v and C_m, the much lower rates of change of C_v and C_m are not sufficient to reduce the power capacity ratings as the centre distance increases from the lower limit. However, for a combination (m_n,N_1,N_2), a small change in centre distance C can cause a significant change in helix angle ψ. See equation (3-2-2). From equation (3-2-20), the change in helix angle ψ will cause a change in face contact ratio m_F, though transverse contact ratio m_p remains more or less the same. As has been shown in the analysis of load sharing ratio m_N, contact ratio derating factor m_{ar} has its minimum value when the sum of face contact ratio m_F and transverse contact ratio m_p is a whole number and m_{ar} has its

Table 4-3 Power capacity ratings versus centre distance

$\lambda=1$, $\psi_{min}=8°$, $\psi_{max}=15°$, $(m_F)_{min}=1.10$

$(m_n, N_1, N_2)=(14,20,84)$

C	ψ	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
739	9.898	1.11	1.60	2365	1171	1681	1221
740	10.332	1.16	1.60	2340	1159	1668	1211
741	10.748	1.21	1.60	2316	1147	1655	1201
742	11.148	1.25	1.60	2296	1137	1644	1193
743	11.532	1.30	1.59	2282	1130	1636	1187
744	11.904	1.34	1.59	2264	1121	1624	1183
745	12.263	1.39	1.59	2254	1116	1620	1179
746	12.612	1.42	1.59	2254	1116	1622	1180
747	12.950	1.46	1.58	2282	1130	1644	1196
748	13.279	1.50	1.58	2318	1148	1671	1215
749	13.600	1.54	1.58	2343	1160	1691	1229
750	13.912	1.57	1.57	2367	1173	1709	1243
751	14.217	1.61	1.57	2397	1187	1731	1262
752	14.514	1.65	1.57	2420	1199	1748	1274
753	14.805	1.68	1.57	2451	1214	1772	1291

(a)

$(m_n, N_1, N_2)=(10,28,119)$

C	ψ	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
743	8.415	1.33	1.69	2219	1099	1179	878
744	8.921	1.41	1.69	2261	1120	1205	898
745	9.398	1.49	1.68	2306	1142	1233	918
746	9.851	1.56	1.68	2339	1159	1254	933
747	10.284	1.63	1.68	2370	1174	1274	947
748	10.698	1.70	1.67	2404	1191	1294	965
749	11.095	1.76	1.67	2430	1203	1310	977
750	11.478	1.82	1.67	2454	1216	1326	988
751	11.848	1.89	1.66	2486	1231	1344	1002
752	12.206	1.94	1.66	2508	1242	1358	1011
753	12.553	2.01	1.66	2533	1255	1373	1022
754	12.890	2.06	1.65	2514	1245	1364	1015
755	13.217	2.11	1.65	2497	1237	1356	1009
756	13.536	2.17	1.65	2485	1231	1350	1004
757	13.847	2.22	1.65	2469	1223	1341	1000
758	14.150	2.27	1.64	2460	1218	1337	997
759	14.447	2.32	1.64	2449	1213	1331	992
760	14.737	2.36	1.64	2440	1209	1326	988

(b)

maximum value when m_F is a whole number. This minimum and maximum value of m_{ar} is passed on to geometry factors I and J and then further on to the power capacity ratings of pitting resistance and bending strength.

Having analysed the behaviour of the power capacity ratings of individual combination (m_n, N_1, N_2), the behaviour of the maximum power capacity rating over a range of centre distance is a compound of the behaviours of individual combinations. Given the basic shape of the curve of the power capacity ratings, the curves of different combinations will intersect on either side of the slopes of the valley or ridge, as shown in Fig.4-13.

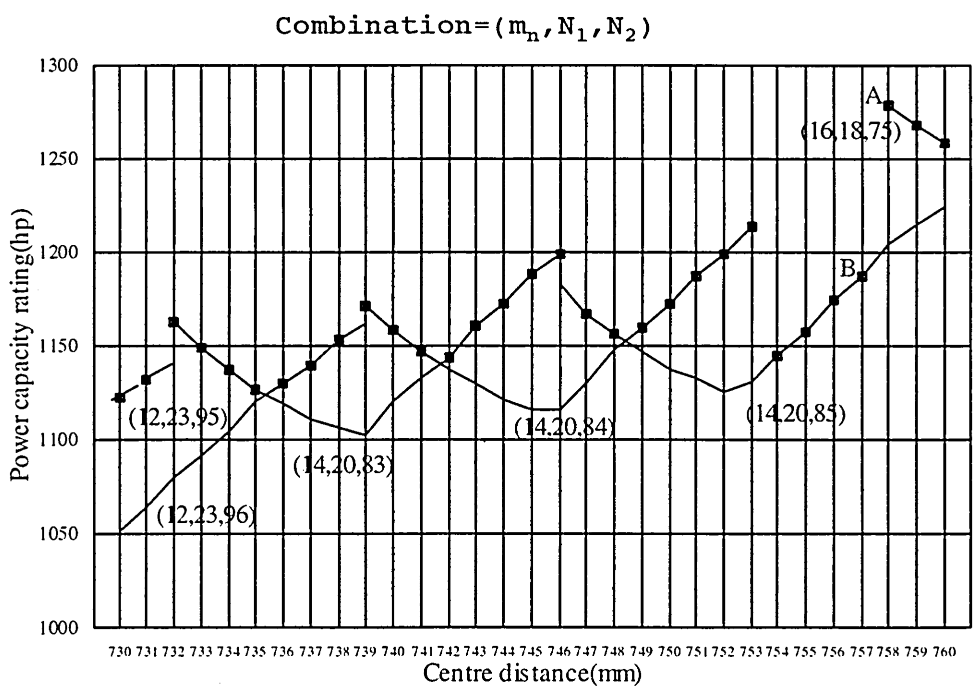


Fig.4-13 Micro non-monotonicity of power capacity rating

The maximum power capacities are shown by the marked points. It can be seen that the step change occurs when the centre distance is changed over the lower or upper limit of the centre distance range of a combination (m_n, N_1, N_2); whereas the lowest point in the valley of the compound curve is the intersecting point of the increasing slope of one combination with the decreasing slope of another.

Thus to eliminate the effect of micro non-monotonicity after the one dimensional search, a local search needs to be carried out to see if there is any smaller centre distance with the maximum power capacity rating greater than the power requirement. When the one dimensional search terminates, the minimised centre distance is at the final upper boundary C^U_{min}. The final lower boundary C^L_{min} is one terminating tolerance smaller than the upper boundary. Thus the local search should start from the final lower boundary C^L_{min} and move towards smaller centre distances.

For brevity in the following discussion, the centre distance that each cycle of local search is started from is called *base centre distance*. The initial base centre distance is the final lower boundary C^L_{min}, which is the centre distance smaller by one terminating tolerance than the current minimised centre distance. *Base combination* is the maximum power combination at the base centre distance. *Reduced centre distance range* starts from the base centre distance to the lower limit of the base combination. Using the example shown in Fig.4-13 and assuming the power requirement is 1200 hp, then point A could be the final upper boundary C^U_{min} and point B the final lower boundary C^L_{min}, if the terminating tolerance is 1 mm. Point B would also be the initial base centre distance and base combination. The reduced centre distance range of the base combination would be from 757 to 746 mm.

To eliminate the effect of the non-monotonicity caused by the valley in the compound power capacity curve, the local search is first to check if there are better combinations in the reduced centre distance range of base combination. Then, to eliminate the effect of the non-monotonicity caused by the step change in the compound power capacity curve, one more centre distance range below the centre distance range of base combination needs to be checked. If no better combinations can be found in the two consecutive reduced centre distance ranges starting from the base centre distance, the current minimised centre distance is declared as the minimum centre distance. Using the example shown in Fig.4-13 again, it is to check the two centre distance ranges from 757 to 746 and from 745 to 732.

To check if there are better combinations in the centre distance range of a combination, only those combinations need to be identified whose power capacity curves intersect with that of the combination to be checked, considering the shape of the compound power capacity curve. Intersection is only possible if the centre distance range of another combination overlaps the reduced centre distance range of the combination to be checked, which means the intersecting combinations must be geometrically feasible at the lower or upper limits of the reduced centre distance range. For these geometrically feasible combinations, power capacity ratings only need to be checked at the lower or upper limits of their centre distance ranges or at the point where face contact ratio m_F is a whole number, since the maximum power capacity rating is only at one of these three points. For the example shown in Fig.4-13, when checking the centre distance range 757 to 746, combination (14,20,84) is geometrically feasible at the lower limit 746, hence the upper limit 753 of combination (14,20,84) is checked to see if its power capacity is larger than the power requirement. Similarly, when checking the centre distance range of 745 to 732, combination (14,20,84) is geometrically feasible at the upper limit 745, hence the lower limit 739 of combination (14,20,84) is checked.

The algorithm for the local search can now be summarised in the following steps:

1. Set the number of failures to 0 of finding better combinations in the reduced centre distance ranges;
2. Store the current minimised centre distance and its maximum power combination;
3. Set the base centre distance(the upper limit of reduced centre distance range) at the above centre distance minus one terminating tolerance;
4. Find all combinations that are geometrically feasible at the base centre distance and identify the maximum power combination as the base combination;
5. Calculate the power capacity ratings at the lower limits of each of these combinations and/or where m_F is a whole number, and store the combination having the largest power capacity;

6. If the largest power capacity rating satisfies the power requirement, return to step 2; otherwise continue;
7. Set the lower limit of the reduced centre distance range by equation (4-5-2), using the base combination;
8. Find all designs that are geometrically feasible at the lower limit;
9. Calculate the power capacity ratings at the upper limits of each of these combinations and/or where m_F is a whole number, and store the combination having the largest power capacity(noting that the upper limits have a cut-off point at the base centre distance);
10. If the largest power capacity rating satisfies the power requirement, return to step 2; otherwise continue;
11. Increase the number of failure by 1;
12. If the number of failure is less than 2, reset the base centre distance(the upper limit of the reduced centre distance range) to the lower limit from step 7 minus one terminating tolerance and return to step 4; otherwise, restore the minimised centre distance and combination stored in step 2 as the minimum centre distance and its maximum power combination.

Thus by eliminating the effect of micro non-monotonicity using the local search algorithm, the minimum centre distance obtained by the one dimensional search is turned from a near global optimum into a global optimum.

4.6 *Numerical Example*

To illustrate how the above algorithm achieves the optimum design, the following example is provided.

The problem is to find the minimum centre distance design (m_n,N_1,N_2,ψ) which satisfies the following requirement:

1.Design specification:

power to be transmitted P = 1200 hp;

input pinion speed n_p = 300 r.p.m., non-reversing;

gear ratio required m_G = 4.2, gear ratio tolerance δ = 2%;

gears are to be single helical type.

2.Gear materials:

pinion carburised and case hardened to HRC 58:

s_{ac1} = 203000 psi, s_{at1} = 56000 psi;

gear through hardened to BHN 320:

s_{ac2} = 133770 psi, s_{at2} = 40480 psi.

3.Factors:

application factor for pitting resistance C_a = 1.75,

application factor for bending strength K_a = 2.00;

life factors C_L = K_L = 1;

reliability factors C_R = K_R = 1.

4.Boundary constraints:

minimum number of pinion teeth $(N_1)_{min}$ = 18;

maximum helix angle ψ_{max} = 15°,

minimum helix angle ψ_{min} = 8°;

minimum overlap ratio $(m_F)_{min}$ = 1.1.

5.Gear tooth system:

normal pressure angle ϕ_n = 20°,

addendum to module ratio h_a = 1,

dedendum to module ratio $h_b = 1.4$,

fillet radius to module ratio $r_b = 0.4$.

6.Other given design requirements:

pinion aspect ratio $\lambda = 1$;

AGMA gear accuracy grade $Q_v = 11$;

prime number larger than 100 cannot be used for the gear;

normal modules m_n available for use(in mm):

4,5,6,7,8,9,10,12,14,16,18,20,22,25,30,34,40.

The design specification and values given above are typical at the collaborating establishment.

The problem is solved by two methods. It is firstly solved by the algorithm developed in this chapter. Secondly, a large range of centre distances is defined, using the minimum centre distance already obtained as a middle point. The power capacities of all possible combinations (m_n, N_1, N_2) at every centre distance in the range are then calculated and a diagram showing the maximum permissible power capacity at every centre distance is drawn, as shown in Fig.4-14. The minimum centre distance is then read directly from the diagram to confirm the optimality of the minimum centre distance found by the algorithm. In the meantime, the optimisation process of the algorithm is explained by using the data created for drawing the diagram.

The minimum centre distance found by the algorithm is 753mm for the combination $(m_n, N_1, N_2)=(14,20,84)$ with helix angle $\psi=14.8054°$. The centre distance range from 700mm to 800mm is defined for confirmation of the optimality. The power capacities of all geometrically feasible combinations are calculated at every centre distance for a step size of 1mm in the range. The results are shown in Appendix B. Fig.4-14 is the maximum power capacity diagram. In the diagram, all the centre distances which have power capacities above the 1200hp line will have at least one feasible design (m_n, N_1, N_2, ψ) satisfying all the geometry and power capacity constraints. Thus the minimum centre distance is the furthest left point above the 1200hp line in the diagram, which is 753mm, the same as that obtained by the algorithm.

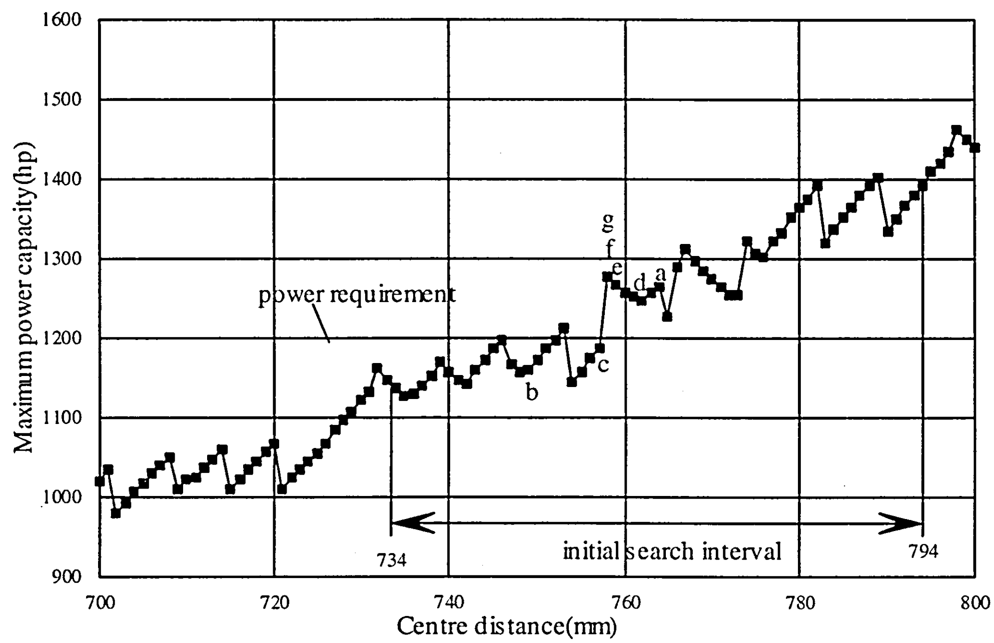


Fig.4-14 One dimensional minimisation

Following is a detailed explanation as to how the algorithm arrives at the minimum centre distance.

The procedure starts with the calculation of the lower and upper boundaries of the search interval for the minimum centre distance. The initial boundaries found by equations (4-2-25), (4-2-26), (4-2-32) and (4-2-33) are C^L_{min}=734 and C^U_{min}=794, as shown in Fig.4-14.

Next is the one dimensional search. The first trial centre distance is the mean of 734 and 794, which is 764, marked as point a in Fig.4-14. Point a is above the power requirement line and hence is used as the new upper boundary. The mean value of 734 and 764 is then calculated as a new trial centre distance, which is 749 and marked as point b. Point b is below the power requirement line and hence is used as the new lower boundary. Repeating the above process, points c,d,e,f and g are used as new trial centre distances and the lower and upper boundaries are updated until the gap between the boundaries is reduced to 1mm. The final lower and upper boundaries resulted from the one dimensional search is 757 and 758, marked as point c and f. Point g of C=758 is the minimum centre distance found by the one dimensional search.

For each of the above trial centre distances, all the geometrically feasible combinations with their power

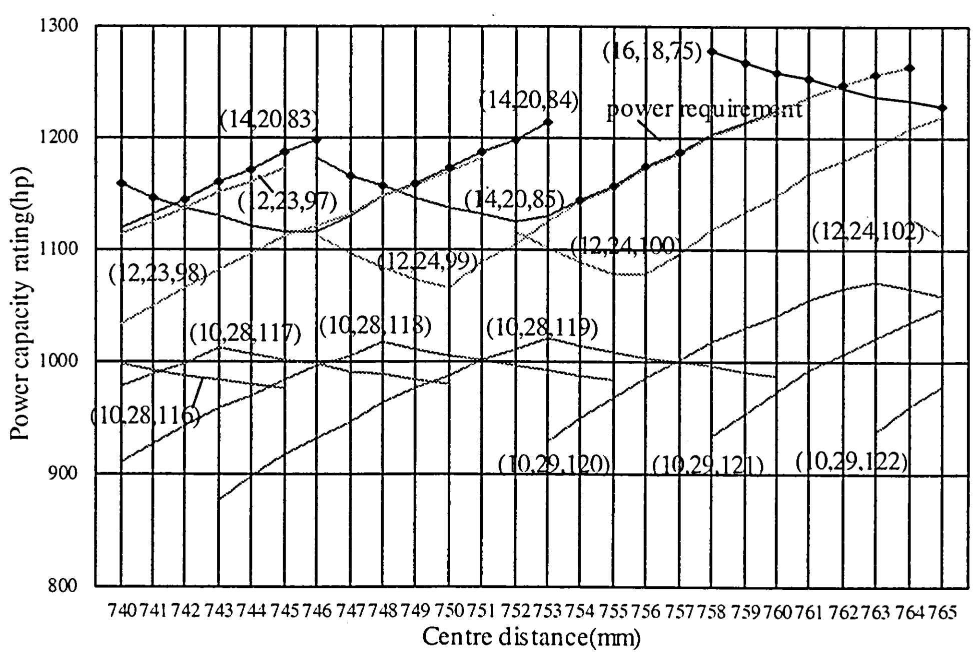


Fig.4-15 Local minimisation of centre distance

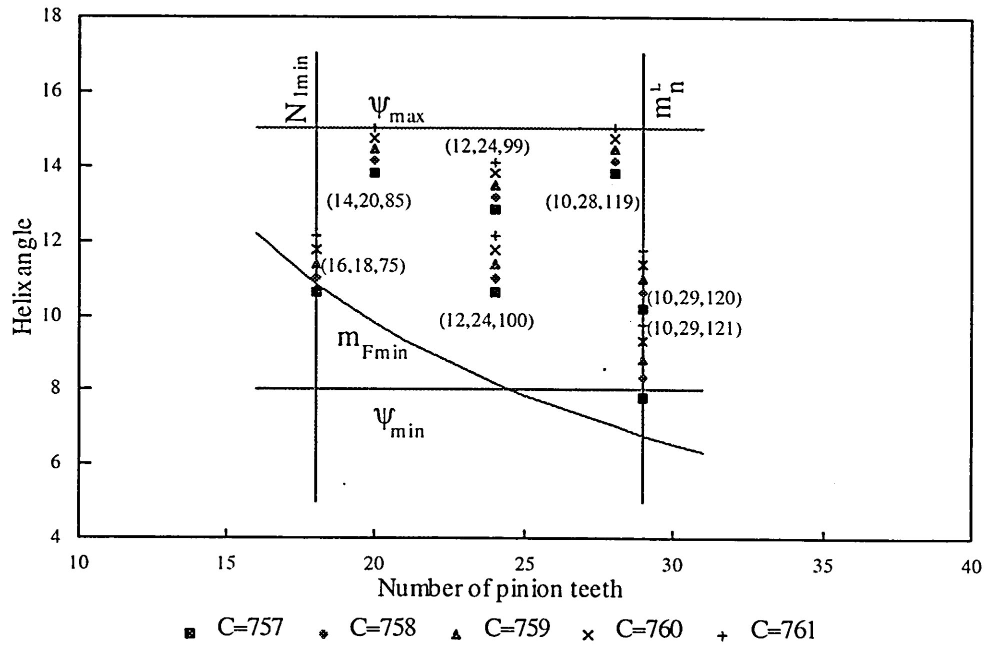


Fig.4-16 Geometrically feasible combinations

capacities are shown in Fig.4-15. Typical geometrical constraints are shown in Fig.4-16 for centre distances from C=757 to C=761. The constraint for gear ratio tolerance is shown separately in Fig.4-17 for C=758. The combination with

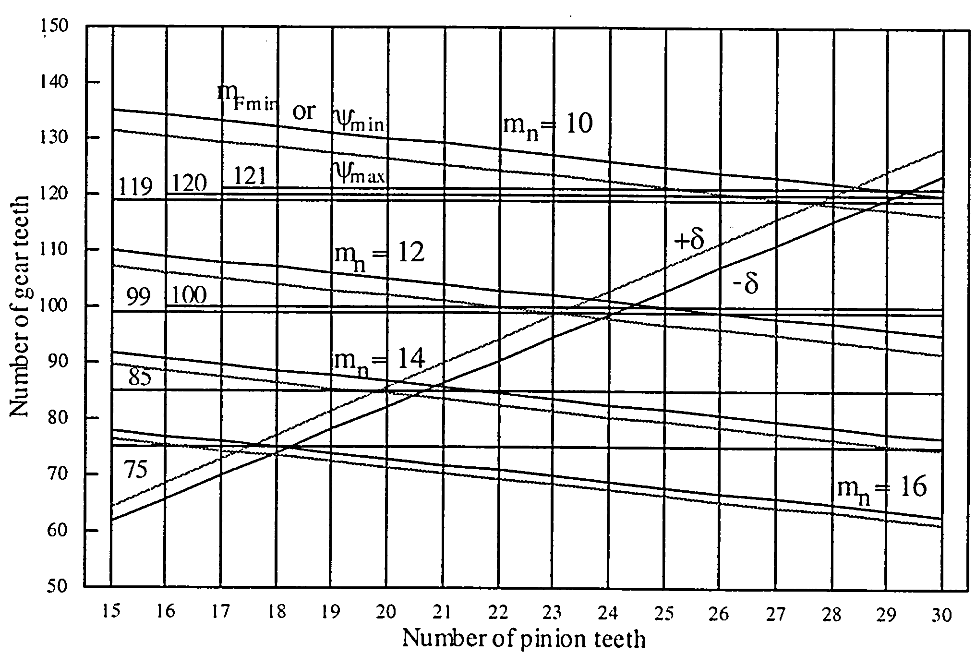


Fig.4-17 Constraints on numbers of teeth

the maximum power capacity at each centre distance is used in the one dimensional search. For centre distances at points a,d,e and f, only one combination (m_n, N_1, N_2) = (16,18,75) is evaluated for its power capacity in the actual optimisation computation. This is because the power capacities of the combination (16,18,75) at these centre distances are above the power requirement line and the module enumeration starts from the upper boundary from equations (4-4-2) and (4-4-4).

Once the one dimensional search has terminated at C=758 in Fig.4-15, the local minimisation process starts from the base centre distance C=757 towards smaller centre distances. The algorithm finds all the geometrically feasible combinations at the base centre distance, which are (14,20,85),(12,24,99), (12,24,100),(10,28,119) and (10,29,120). The combination (12,24,101) has been excluded since 101 is a prime number. The power capacities at the base centre distance are evaluated and combination (14,20,85) is the base combination with the maximum power capacity. The power capacities of these combinations at their lower limits or where m_F is a whole number are then evaluated, which are at centre distances 746 for (14,20,85), 746 for (12,24,99), 752 for (12,24,100), 753 for (10,28,119) and 753 for (10,29,120). None of these points are above the power requirement line. The algorithm then finds all the geometrically feasible combinations at C=746, the lower limit of the base combination (14,20,85). These combinations are (14,20,83), (14,20,85), (12,23,98), (14,20,84) and (12,24,99). The combinations (10,28,117), (10,28,118) and (10,28,119) are excluded because the lower boundary on the module at C=746 from equation (4-4-8) is C^L_{min}=12. The power capacities of these combinations at their upper limits or where m_F is a whole number are then evaluated, which are at centre distances 746 for (14,20,83), 757 for (14,20,85), 751 for (12,23,98), 753 for (14,20,84), and 757 for (12,24,99). Of these points, the combination (14,20,84) at C=753 is above the power requirement line and hence C=753 is the new minimum centre distance. A new round of local minimisation is then started from the new base centre distance C=752. After two more rounds in failing to find smaller centre distances that satisfy the power requirement, centre distance C=753 and combination (14,20,84) is declared the minimum centre distance design. The helix angle is calculated by using equation (3-2-2) and is ψ=14.8054°.

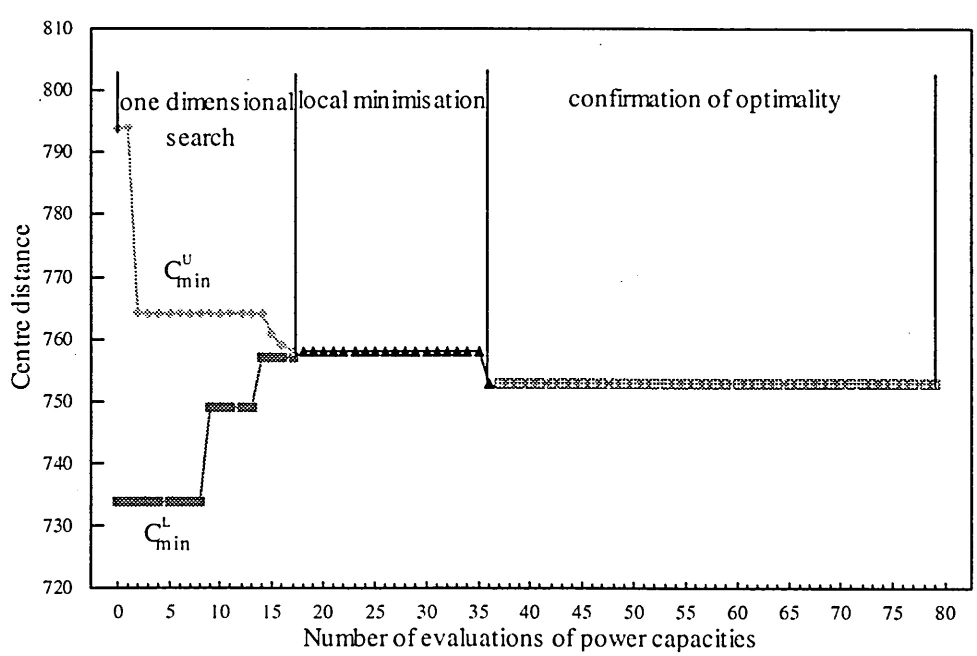


Fig.4-18 Iteration efficiency

The optimality of other minimum centre distances found by the algorithm for other power requirements is also confirmed at C=789 for P=1400 hp, C=767 for P=1300 hp, and C=729 for P=1100 hp. The efficiency of the algorithm for the example problem is shown in Fig.4-18. Obviously, the one dimensional search is very efficient to arrive at C=758 with only 17 evaluations of the power capacities. It then takes 19 more evaluations to arrive at the new minimum centre distance C=753 and another 43 evaluations to confirm that C=753 is the global minimum centre distance.

CHAPTER FIVE GEAR SHAFT DESIGN

5.1 *Descriptive Layout Definition*

In order to calculate bearing reactions of a gear shaft, the layout of a gear box has to be defined. In addition, the layout is used to check the possibility of interferences between bearing outside diameters, gear tip circles and shaft outside diameters.

Fig.5-1a & 5-1b are two examples of gear box layouts. Fig.5-1a is a twin drive, i.e., a gear box with two inputs and two outputs. Fig.5-1b is a plain reduction box with the output shaft extended to take the moment caused by a heavy external load applied at the output coupling. To deal with complicated layouts such as that of Fig.5-1a and be universally applicable to different layouts, it is decided that a descriptive definition of layout is to be used and is devised as follows.

The layout definition is based on vectors connecting the gear centres in the Cartesian coordinate system. The Z axis must always be parallel to the shaft axes. By convention, the positive direction of Z axis goes from the input side to the other side, but it can take the opposite direction if necessary. A datum point need to be chosen for the Z coordinate for axial position references. The input side bearing centre can often be chosen as the datum point. The X axis can be chosen for convenience. For instance, It can be horizontal or goes from the first reduction pinion centre to the first reduction gear centre. The Y axis must follow the right hand rule with the X and Z axes. Once X and Z axes have been chosen, Y axis is fixed.

When the coordinate system has been established, the layout is defined by vectors connecting gear centres and axial

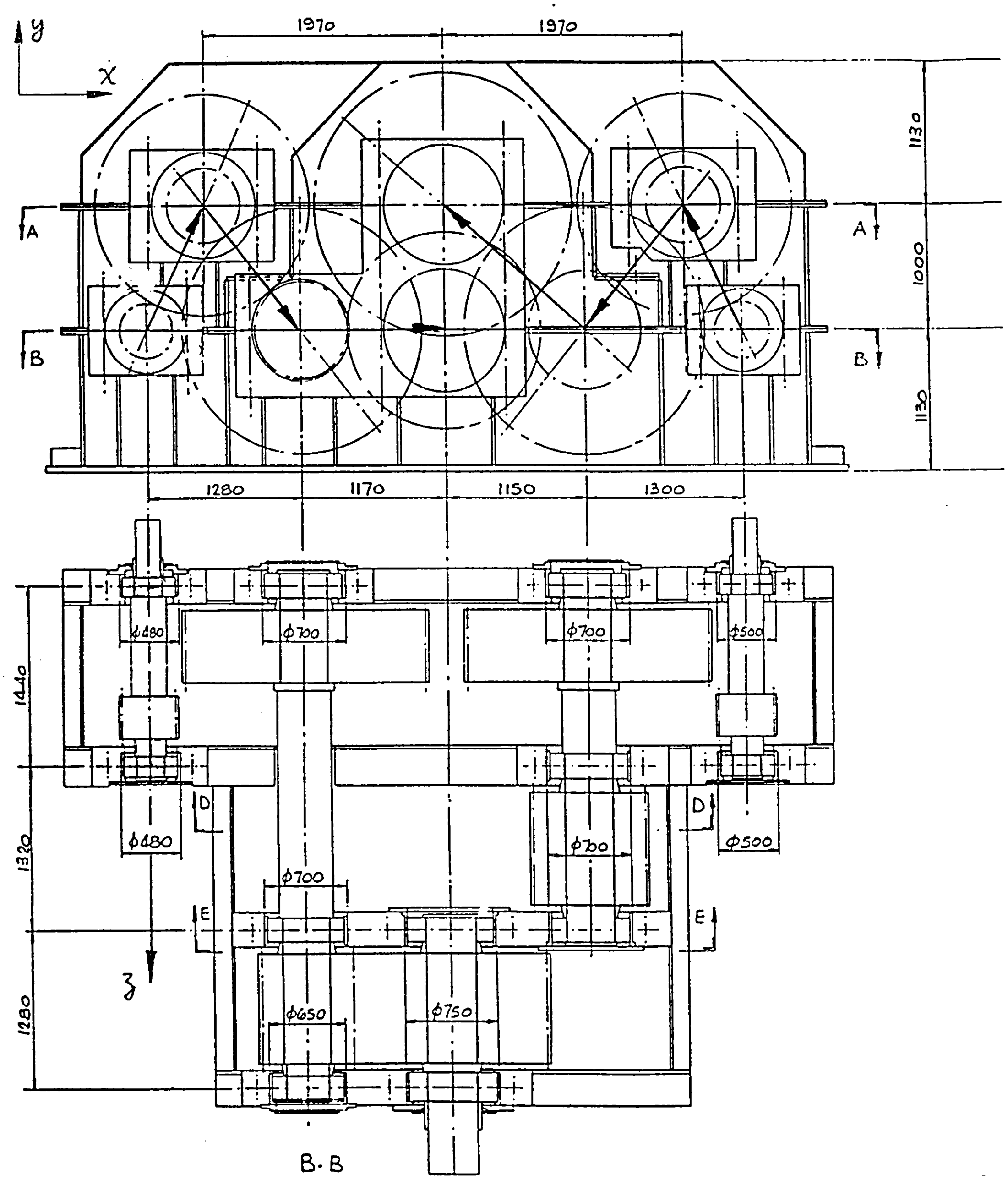


Fig.5-1a Gear Box Layout Definition

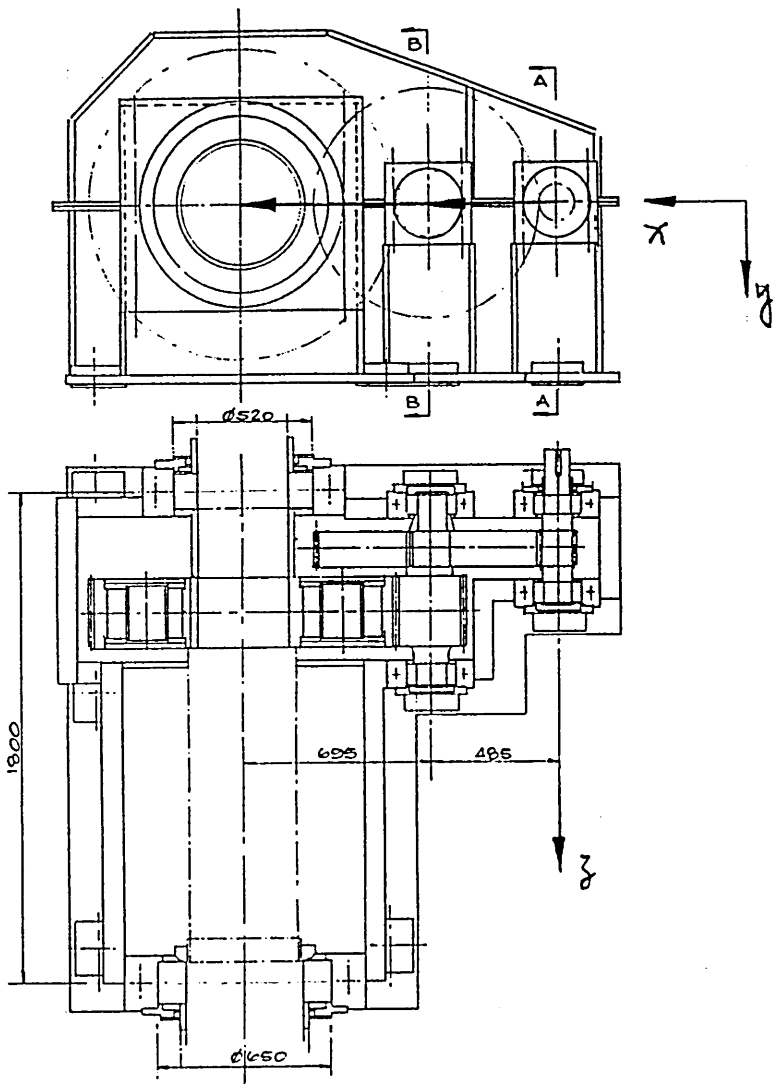


Fig.5-1b Gear Box Layout Definition

positions of each and every part and load. The vectors are referred to as centre vectors. They always go from the driver to the driven gear, or pinion to gear for reduction gears. The magnitude of the vector is the centre distance, which is known from the gear particulars. The direction of the centre vector can be specified in one of the following three ways:

1. Directional angle: the directional angle is the angle between the positive X and the centre vector, measured in the direction rotating from positive X to positive Y through the smaller angle between X and Y.
2. X component and Y bias: when the X component of the centre vector is specified, the magnitude of the Y component can be worked out from the known centre distance, hence only the bias of the vector(to the positive Y or negative Y) is required to define the centre vector.
3. Y component and X bias: when the Y component of the centre vector is specified, the magnitude of the X component can be worked out from the known centre distance, hence only the bias of the vector(to the positive X or negative X) is required to define the centre vector.

Take the layout of Fig.5-1a for example. The coordinate system is defined as shown. For the left hand side gear train, the first reduction centre vector can be defined by specifying Y component as being 1000 and positive X bias. The second reduction centre vector can be defined by specifying Y component as being -1000 and positive X bias. The third reduction centre vector can be defined by specifying the directional angle as being 0. Similarly, for the right hand side gear train, the first reduction centre vector has a Y component of 1000 and biases to negative X. The second reduction centre vector has a Y component of -1000 and biases to negative X. The third reduction centre vector has a Y component of 1000 and biases to negative X. The axial positions of all the gears, bearings, external loads and torques on each and every shaft must also be specified to complete the definition of the layout of a gear box.

5.2 *Gear Shaft Load Analysis*

The loads on a gear shaft come from the force created by gear mesh, external torques and loads such as those applied by coupling, and bearing reactions. The force created by gear mesh is normal to the tooth surface at the pitch point. The position of the pitch point relative to the gear shaft is determined from the directional angle of the centre vector, the radius of pitch circle and the axial position of the gear. The gear mesh force is decomposed into three mutually perpendicular components as shown in Fig.5-2: one is tangent

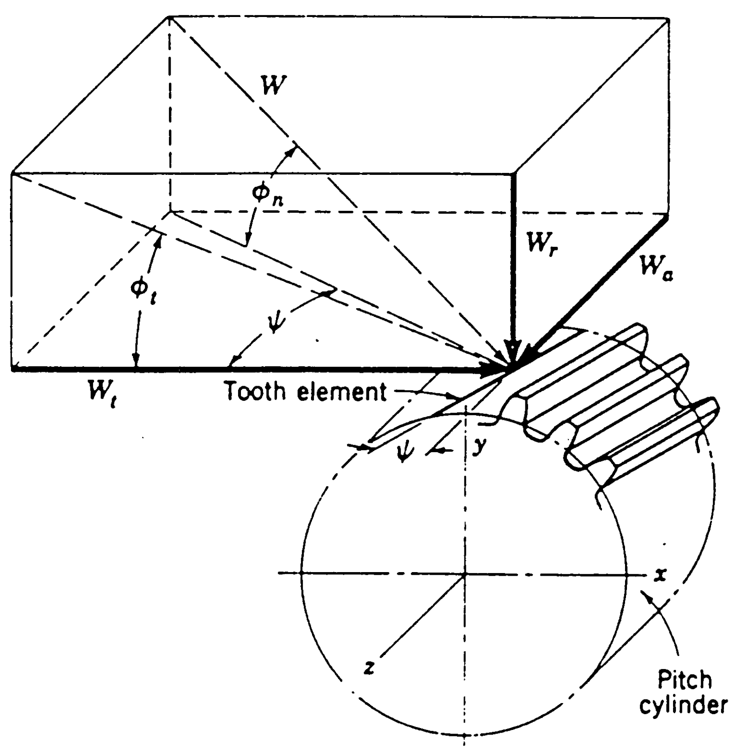


Fig.5-2 Force Components on Helical Gear

to the pitch circles at the pitch point and is called tangential load W_t; one is along the pitch point radial direction and is called radial load W_r; and one is parallel to the axes of the gears and is called axial load W_a. Note that all three components are applied at the pitch point and can be calculated by the following equations.

$$W_t = \frac{126000\ P}{n_p\ d} \tag{5-2-1}$$

$$W_r = W_t \frac{\tan\phi_n}{\cos\psi} \tag{5-2-2}$$

$$W_a = W_t \tan\psi \tag{5-2-3}$$

Once the gear loads have been calculated, the only unknown loads are the bearing reactions, since external loads are normally given.

Reaction calculation of a determinate structure problem such as a shaft supported by two bearings is simple. Reaction calculation of an indeterminate structure problem such as a shaft supported by three bearings can be fairly difficult, especially if the shaft has different section diameters along its axis. The latter case is typical for the output mill pinion shaft of a compact drive at the collaborating establishment. To deal with reaction calculation of both cases, a finite element model for beams given in Stasa [48] is adopted. In this model, the finite element characteristics for a beam element are based on simple plane bending theory. The method can be summarised in the following steps:

1. Discretization.

The shaft is discretized into segments of shaft journals. Each segment is an element and each section separating two adjacent segments is a node. Wherever there is a load, a bearing, or a change of section size, there should be a node so that within each element, there is no external load or change of section size.

2. Determination of element characteristics.

The relationship of the element stiffness matrix, the nodal force vector and the nodal displacement vector of an element e can be expressed by

$$K^e a^e = f^e \tag{5-2-4}$$

K^e is the element stiffness matrix, given by

$$K^e = \frac{EI_z}{L_e^3} \begin{bmatrix} 12 & 6L_e & -12 & 6L_e \\ 6L_e & 4L_e^2 & -6L_e & 2L_e^2 \\ -12 & -6L_e & 12 & -6L_e \\ 6L_e & 2L_e^2 & -6L_e & 4L_e^2 \end{bmatrix} \tag{5-2-5}$$

where

E = modulus of elasticity

I_z = moment of inertia of the cross section

L_e = length of the element(shaft segment)

a^e is the displacement vector, given by

$$a^e = \begin{bmatrix} w_i \\ 0_i \\ w_j \\ 0_j \end{bmatrix} \tag{5-2-6}$$

where

w = nodal deflection

0 = nodal slope

i = left node index

j = right node index

f^e is the force vector, given by

$$f^e = \begin{bmatrix} -V_i \\ -M_i \\ +V_j \\ +M_j \end{bmatrix} \qquad (5\text{-}2\text{-}7)$$

where

V = nodal shear force

M = nodal bending moment

The sign conventions are as shown in Fig.5-3 . Note K^e is a symmetric matrix.

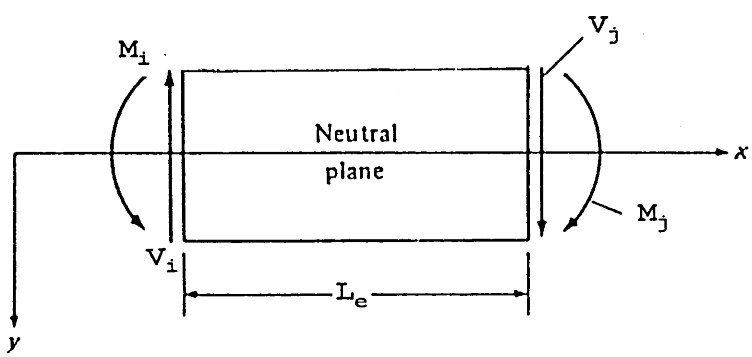


Fig.5-3 Sign Conventions for a Beam Element

3. Assemblage of global element characteristics

Only adjacent elements have a common node. For the assemblage of stiffness matrix, this means only the bottom right two rows and two columns of the previous element stiffness matrix will be amalgamated with the top left two rows and two columns of the next element stiffness matrix, as shown in Fig.5-4. The size of the assembled stiffness matrix is $2N_n$, where N_n is the number of nodes. The band width of the assembled stiffness matrix is 6. For the assemblage of force vectors, any external force or couple applied at a node is to be considered by one element only.

#	#	#	#	0	0	0	0	node-	
#	#	#	#	0	0	0	0	1	element 1
#	#	+	+	#	#	0	0	node-	
#	#	+	+	#	#	0	0	2	element 2
0	0	#	#	+	+	#	#	node-	
0	0	#	#	+	+	#	#	3	element 3
0	0	0	0	#	#	#	#	node-	
0	0	0	0	#	#	#	#	4	

= original element stiffness value unmodified

+ = element stiffness values amalgamated

Fig. 5-4. Assembled stiffness matrix pattern

4. Application of restraints on displacement at nodes.

After the assemblage of stiffness matrices and force vectors, the finite element equation system is given by

$$\begin{bmatrix} k_{11} & k_{12} & \dots & k_{1i} & \dots & k_{1n} \\ k_{21} & k_{22} & \dots & k_{2i} & \dots & k_{2n} \\ . & . & & . & & . \\ k_{i1} & k_{i2} & \dots & k_{ii} & \dots & k_{in} \\ . & . & & . & & . \\ k_{n1} & k_{n2} & \dots & k_{ni} & \dots & k_{nn} \end{bmatrix} \begin{Bmatrix} a_1 \\ a_2 \\ . \\ a_i \\ . \\ a_n \end{Bmatrix} = \begin{Bmatrix} f_1 \\ f_2 \\ . \\ f_i \\ . \\ f_n \end{Bmatrix} \qquad (5\text{-}2\text{-}8a)$$

or simply

$$K\ a = f \qquad (5\text{-}2\text{-}8b)$$

where K is the stiffness matrix, a is the displacement vector and f is the force vector. A restraint on displacement a_i can be applied by making a_i equal to the prescribed value, say, a_p. Thus the finite element equations can be re-written as

$$\begin{bmatrix} k_{11} & k_{12} & \dots & 0 & \dots & k_{1n} \\ k_{21} & k_{22} & \dots & 0 & \dots & k_{2n} \\ . & . & & . & & . \\ 0 & 0 & 0 & 1 & 0 & 0 \\ . & . & & . & & . \\ k_{n1} & k_{n2} & \dots & 0 & \dots & k_{nn} \end{bmatrix} \begin{Bmatrix} a_1 \\ a_2 \\ . \\ a_i \\ . \\ a_n \end{Bmatrix} = \begin{Bmatrix} f_1 - k_{1i}a_p \\ f_2 - k_{2i}a_p \\ . \\ a_p \\ . \\ f_n - k_{ni}a_p \end{Bmatrix} \qquad (5\text{-}2\text{-}9)$$

The ith column is transformed as is the ith row to keep the symmetry of the stiffness matrix. For a bearing, it is a restraint on deflection and $a_p=0$.

5. Solution of the finite element equations.

The finite element equations of a shaft can be solved for the displacements by LU(triangular) decomposition, forward elimination and backward substitution. The stiffness matrix can be decomposed into two matrices:

$$\begin{bmatrix} k_{11} & k_{12} & \dots & k_{1n} \\ k_{21} & k_{22} & \dots & k_{2n} \\ \cdot & \cdot & & \cdot \\ k_{n1} & k_{n2} & \dots & k_{nn} \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ L_{21} & 1 & 0 & 0 \\ \cdot & & 1 & 0 \\ L_{n1} & L_{n2} & \dots & 1 \end{bmatrix} \begin{bmatrix} U_{11} & U_{12} & \dots & U_{1n} \\ 0 & U_{22} & \dots & U_{2n} \\ 0 & 0 & \cdot & \cdot \\ 0 & 0 & 0 & U_{nn} \end{bmatrix}$$

or simply

$$K = LU \tag{5-2-10}$$

So the finite element equations can now be written as

$$LUa = f \tag{5-2-11}$$

Let vector $y = Ua$, then

$$Ly = f \tag{5-2-12}$$

which can be solved by forward elimination, starting from $y_1 = f_1$. After obtaining vector y, a can be solved for by

$$Ua = y \tag{5-2-13}$$

using backward substitution, starting from $a_n = y_n/U_{nn}$.

6. Bearing reaction calculations.

Once the displacements have been obtained, the stiffness vectors for shear force at the nodes of bearings are used with the displacement vector to obtain the reactions at the bearings.

5.3 *Bearing Selection*

Anti-friction or rolling contact bearings are used for the support of shafts in most gear boxes. Different types of bearings are manufactured to take pure radial loads, pure thrust loads, or a combination of these two. Because of the limited time available for this investigation, only one type of bearing is considered, which is the spherical roller bearing, made of two rows of barrel shaped rollers arranged in an arch within the bearing width, as shown in Fig.5-5. This type of bearing can take both radial and thrust loads and can accommodate some angle misalignment in shaft and bearing assembly. It is used in the majority of the gear boxes manufactured by the collaborating establishment.

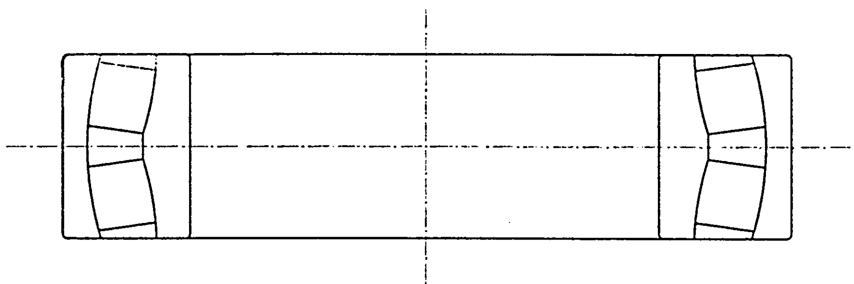


Fig.5-5 Spherical Roller Bearing

Within each bearing type, bearings are grouped into series by width and outside diameter. Five series of spherical roller bearings are most often used at the collaborating establishment. They are 230**, 240**, 231**, 241**, and 232**. In these designations, the first digit 2 represents the spherical roller bearing; the second digit represents the width series, the larger number the wider; the third digit represents the outside diameter series, the larger number the bigger; and the asterisks represent the bore size.

The primary factor in the selection of bearing size is the life consideration. Sometimes a minimum bore may be required because of, for instance, coupling size. Bearing life in millions of revolutions is expressed by [46]:

$$L_{10} = \left(\frac{C_1}{F_e}\right)^p \tag{5-3-1}$$

where

L_{10} = life, in millions of revolutions, probability of failure within life less than 10%;

C_1 = dynamic capacity rating, N, at 1 million revolutions;

F_e = equivalent dynamic load, N;

p = 10/3 for roller bearings(3 for ball bearings).

The dynamic capacity rating C_1 of any particular bearing can be found in bearing manufacturers' catalogues, such as the SKF *General Catalogue* [47]. The equivalent dynamic load of a combination of radial and thrust loads is calculated by [46]

$$F_e = XF_r + YF_a \tag{5-3-2}$$

where

F_r = radial load, N

F_a = thrust load, N

X = radial factor

Y = thrust factor

For SKF spherical roller bearings,

$X = 1.00$ and $Y = Y_1$ when $F_a/F_r \le e$

$X = 0.67$ and $Y = Y_2$ when $F_a/F_r > e$

where Y_1, Y_2 and e are found in [47]. Since C_1, Y_1, Y_2 and e are dependent on the bearing selected, equations (5-3-1) and (5-3-2) can only be used for bearing life rating.

Bearing selection is a reverse problem. The required bearing life is given, the radial and thrust loads on the bearing are known from the gear shaft reaction calculations. The problem is to select a bearing as small as possible that will take the loads and meet the life requirement. The difficulty is in that C_1, Y_1, Y_2 and e are unknown before the bearing is selected. By analysing the bearing data from [47], it can be found that Y_1, Y_2 and e are approximately correlated as follows:

$$Y_2 \approx \frac{1}{e} \tag{5-3-3}$$

$$Y_1 \approx \frac{2}{3} Y_2 \tag{5-3-4}$$

Each series has a relatively narrow band of e values and the equivalent dynamic load is as shown in Fig.5-6, for the example of series 230**. The dynamic capacity rating C_1 of each series is correlated to the bore in the following form,

$$C_1 = ad^2 + bd + c \tag{5-3-5}$$

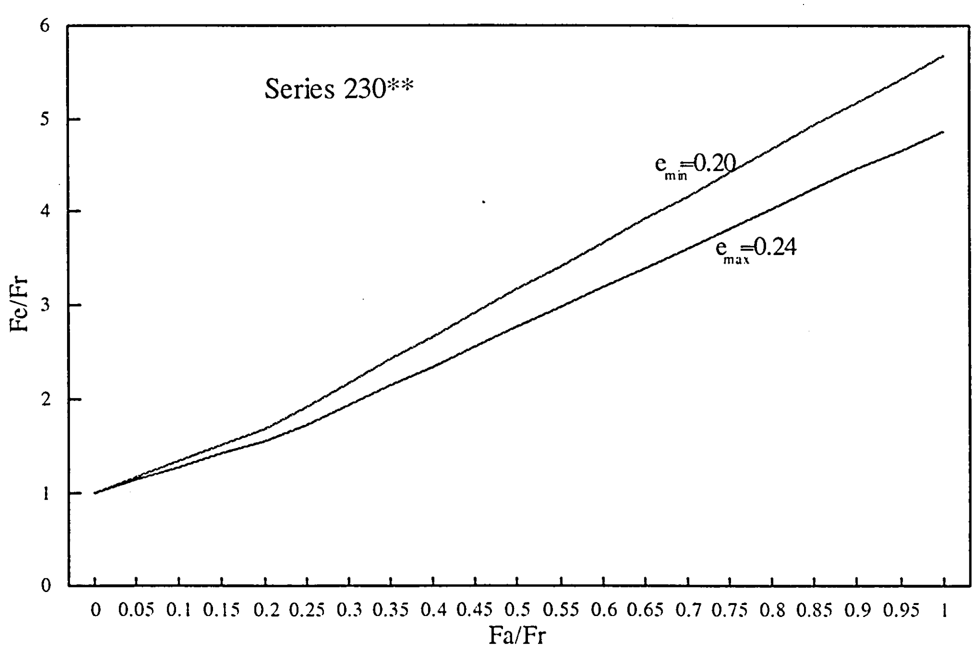


Fig.5-6 Equivalent dynamic load

where

d = diameter of the bearing bore, mm

a, b and c = correlating coefficients and constant

Using the data from [47], the correlating coefficients and constant a,b and c for each series are found and listed in Table 5-1. The accuracy of an estimate using equation (5-3-5) and Table 5-1 can be seen in Fig.5-7, where the symbolic marks are catalogue values.

Table 5-1 Dynamic capacity as a function of bore Coefficients and Constant

	230**	240**	231**	241**	232**
a	0.008867	0.009645	0.018909	0.022396	0.037318
b	4.174261	6.610066	4.402126	6.106021	-0.6192
c	-329.037	-619.417	-337.593	-570.89	191.0248

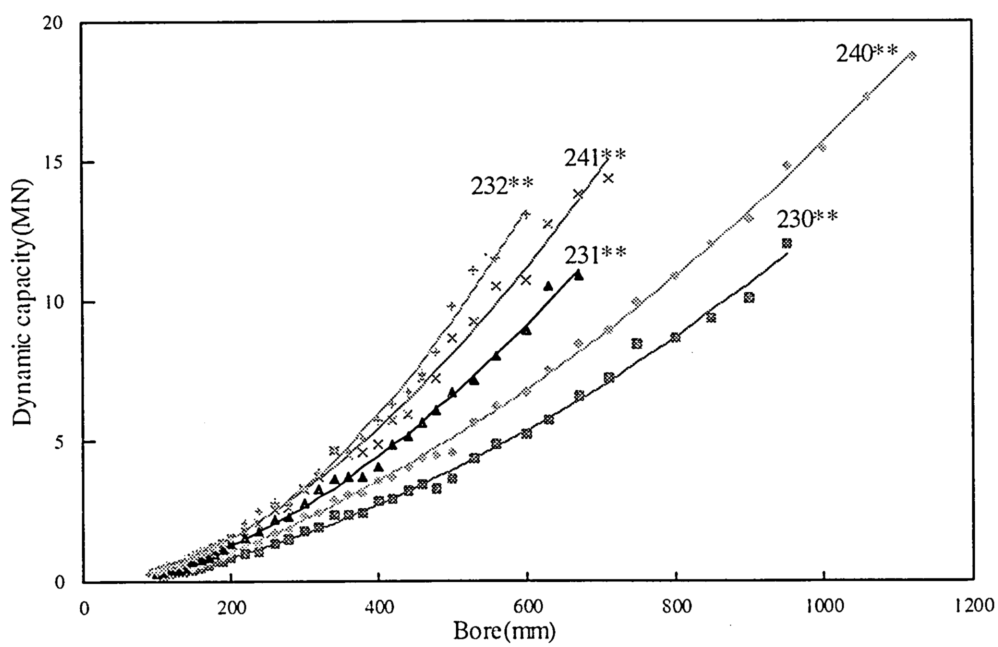


Fig.5-7 Dynamic capacity as a function of bore

Having established the above equations, bearings can be selected efficiently by the following procedure.

1. The maximum value of e and its corresponding Y_1 and Y_2 are used to calculate the equivalent load F_e.
2. The required dynamic capacity C_1 is solved for from equation (5-3-1) and is calculated by

$$C_1 = \left(\frac{60nL_{10h}}{10^6}\right)^{1/p} F_e \qquad (5\text{-}3\text{-}6)$$

where

n = bearing speed in r.p.m.

L_{10h} = bearing life in hours

3. Equation (5-3-5) and Table 5-1 are used to solve for the bearing bore:

$$d = \frac{-b + [b^2 - 4a(c-C_1)]^{1/2}}{2a} \qquad (5\text{-}3\text{-}7)$$

The bearing bore so obtained using the maximum value of e is the lower boundary of bore.

4. The smallest catalogue bore size larger than the value given by equation (5-3-7) and the required minimum bore, if specified, is found.
5. The bore two sizes smaller than the bore found in step 4 but not smaller than the minimum required bore is used to eliminate the effect of any discrepancy between the estimated dynamic capacity and the actual value from the catalogue.
6. The bearing life with the bore size from step 5 is checked, using C_1, e, Y_1 and Y_2 from the catalogue for that bore size.

7. If the life is longer than the required value, the bearing with this bore size is selected; otherwise, the life of the bearing with the next larger size bore is checked and repeated if necessary until the life requirement is satisfied.
8. The above procedure is repeated for each series.
9. The bearing of the series with the smallest bore is selected. If more than one series have the same smallest bore, the lighter series is selected. Series 230**, 240**, 231**, 241** and 232** are in ascending order for their weights.

5.4 *Shaft Diameter Design*

Shafts in gear boxes are normally stepped shafts with diameters for coupling, gear wheel, and bearings. Stress levels in the shaft diameters for coupling and gear wheel are often the determining factors for the diameters. Deflection of the shaft, especially over the face width, is normally checked after the shaft diameters have been determined. For high speed boxes, critical speed may also need to be checked.

The stresses in a rotating gear shaft are torsional stress and bending stress. The torsional stress is considered as a steady shear stress if the starts and stops are few. For more frequent starts and stops, the torsional stress should be considered as pulsating stress with the number of starts and stops counted as the number of stress cycles. For frequent reversing rotations, the torsional stress should be considered as alternating stress. The bending stress is always treated as alternating stress due to the rotation of the shaft, with every revolution counted as one stress cycle. So the number of bending stress cycles easily exceeds the threshold of 10^6 or 10^7 for infinite life in fatigue analysis. ANSI/ASME B106.1M [10] gives the following design formula, based on the distortion energy or von Mises-Hencky failure theory, for determining the shaft diameter of steady torsional stress and alternating bending stress.

$$d = \left\{\frac{32F_S}{\pi}\left[\left(\frac{M}{S_e}\right)^2 + \frac{3}{4}\left(\frac{T}{S_y}\right)^2\right]^{1/2}\right\}^{1/3} \qquad (5\text{-}4\text{-}1)$$

where

d = shaft diameter, m

F_S = factor of safety

M = bending moment, Nm

T = mean static torque, Nm

S_e = tensile endurance limit, N/m^2

S_y = tensile yield strength, N/m^2

At the collaborating establishment, a more conservative approach is taken [22] by treating torsional stress as pulsating fatigue stress for unidirectional rotation or alternating fatigue stress for reversing rotation. For unidirectional rotation, the alternating part τ_a of the pulsating torsional stress is

$$\tau_a = \frac{8T}{\pi d^3} \qquad (5\text{-}4\text{-}2)$$

and the mean part τ_m of the pulsating torsional stress is

$$\tau_m = \frac{8T}{\pi d^3} \qquad (5\text{-}4\text{-}3)$$

For reversing rotation, the alternating torsional stress τ_a is

$$\tau_a = \frac{16T}{\pi d^3} \tag{5-4-4}$$

The alternating bending stress σ_a for both unidirectional and reversing rotation is

$$\sigma_a = \frac{32M}{\pi d^3} \tag{5-4-5}$$

The shaft diameters are so determined that the stresses and allowable stresses satisfy the following elliptical relation

$$R_1^2 + R_2^2 = 1 \tag{5-4-6}$$

where R_1 and R_2 are stress ratios for bending and torsional stresses, defined as

$$R_{1,2} = \frac{\text{actual stress}}{\text{allowable stress}} \tag{5-4-7}$$

For the alternating bending stress, the stress ratio is

$$R_1 = \frac{\sigma_a}{S_e} \tag{5-4-8}$$

For the alternating torsional stress of reversing rotation, the stress ratio is

$$R_2 = \frac{\tau_a}{S_{se}} \tag{5-4-9}$$

where S_{se} is the endurance limit for shear stress and is

$$S_{se} = \frac{S_e}{3^{1/2}} \tag{5-4-10}$$

For the situation of unidirectional rotation, a modified Goodman diagram, as shown in Fig.5-8, is used for the calculation of the torsional stress ratio:

$$R_2 = \frac{\tau_a}{S_{se}} + \frac{2}{3}\frac{\tau_m}{S_{sy}} \tag{5-4-11}$$

where S_{sy} is the shear yield strength and is

$$S_{sy} = \frac{S_y}{3^{1/2}} \tag{5-4-12}$$

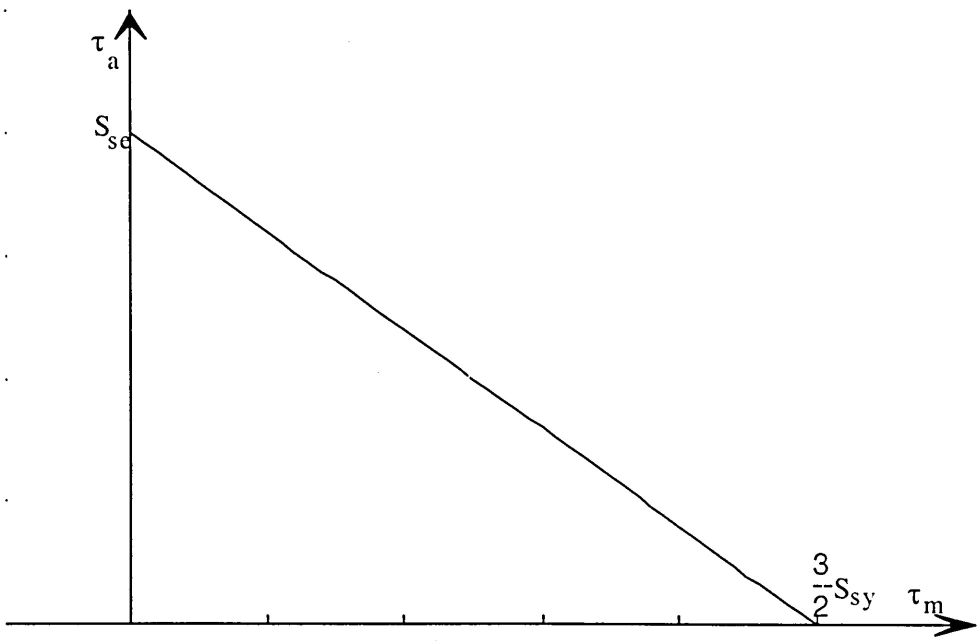


Fig.5-8 Modified Goodman Diagram

Considering a factor of safety F_S, the elliptical relation for the reversing rotation is hence

$$\left(\frac{F_S\sigma_a}{S_e}\right)^2 + \left(\frac{F_S\tau_a}{S_{se}}\right)^2 = 1 \tag{5-4-13}$$

Substituting equations (5-4-4) for τ_a and (5-4-5) σ_a, equation (5-4-13) can be solved for d in circumstances of reversing rotation:

$$d = \left\{\frac{32F_S}{\pi}\left[\left(\frac{M}{S_e}\right)^2 + \frac{1}{4}\left(\frac{T}{S_{se}}\right)^2\right]^{1/2}\right\}^{1/3} \qquad (5\text{-}4\text{-}14)$$

Noticing equation (5-4-10), equation (5-4-14) is similar to (5-4-1). For unidirectional rotation, the elliptical relation with the safety factor F_S is

$$\left(\frac{F_S\sigma_a}{S_e}\right)^2 + \left(\frac{F_S\tau_a}{S_{se}} + \frac{2}{3}\frac{\tau_m}{S_{sy}}\right)^2 = 1 \qquad (5\text{-}4\text{-}15)$$

Substituting equations (5-4-2) and (5-4-3) for τ_a and τ_m, and (5-4-5) for σ_a, equation (5-4-15) can be solved for d in circumstances of unidirectional rotation:

$$d = \left\{\frac{32F_S}{\pi}\left[\left(\frac{M}{S_e}\right)^2 + \frac{T^2}{16}\left(\frac{1}{S_{se}} + \frac{2}{3S_{sy}}\right)^2\right]^{1/2}\right\}^{1/3} \qquad (5\text{-}4\text{-}16)$$

6.1 *Program Structure and Logical Flow*

The gear design method developed in Chapter Four and the shaft design method developed in Chapter Five are implemented in the software package DRIVES. The architecture of the package is a parallel structure of functional branches. Each functional branch can be used independently or in conjunction with others. There are four of them, namely, GEAR DESIGN, GEAR RATING, SHAFT DESIGN and SHAFT CHECK. GEAR RATING and SHAFT CHECK are used to check the ratings and stress levels of existing designs, while GEAR DESIGN and SHAFT DESIGN are used to produce new designs from a minimum amount of information in the specification. The overall structure of the DRIVES package has been shown in Fig.1-3 which is reproduced as Fig.6-1. The programs are written in VAX Fortran [23] & [24] and are organised into 284 modules. A list of the name and purpose of each module in alphabetical order is given in Appendix C. The total number of source program lines is 23351. Due to the limited space of this thesis, it is only possible to show here the program calling structure of the package. The program calling structure is basically a tree type structure, with the module DRIVES as the root of the tree and four main branches of GEAR_DESIGN, SHAFT_DESIGN, GEAR_RATING and SHAFT_CHECK. Thus the calling structure is shown in five groups, one for the trunk and four for the branches. Detailed expansion of the program calling structure is provided in appendix D.

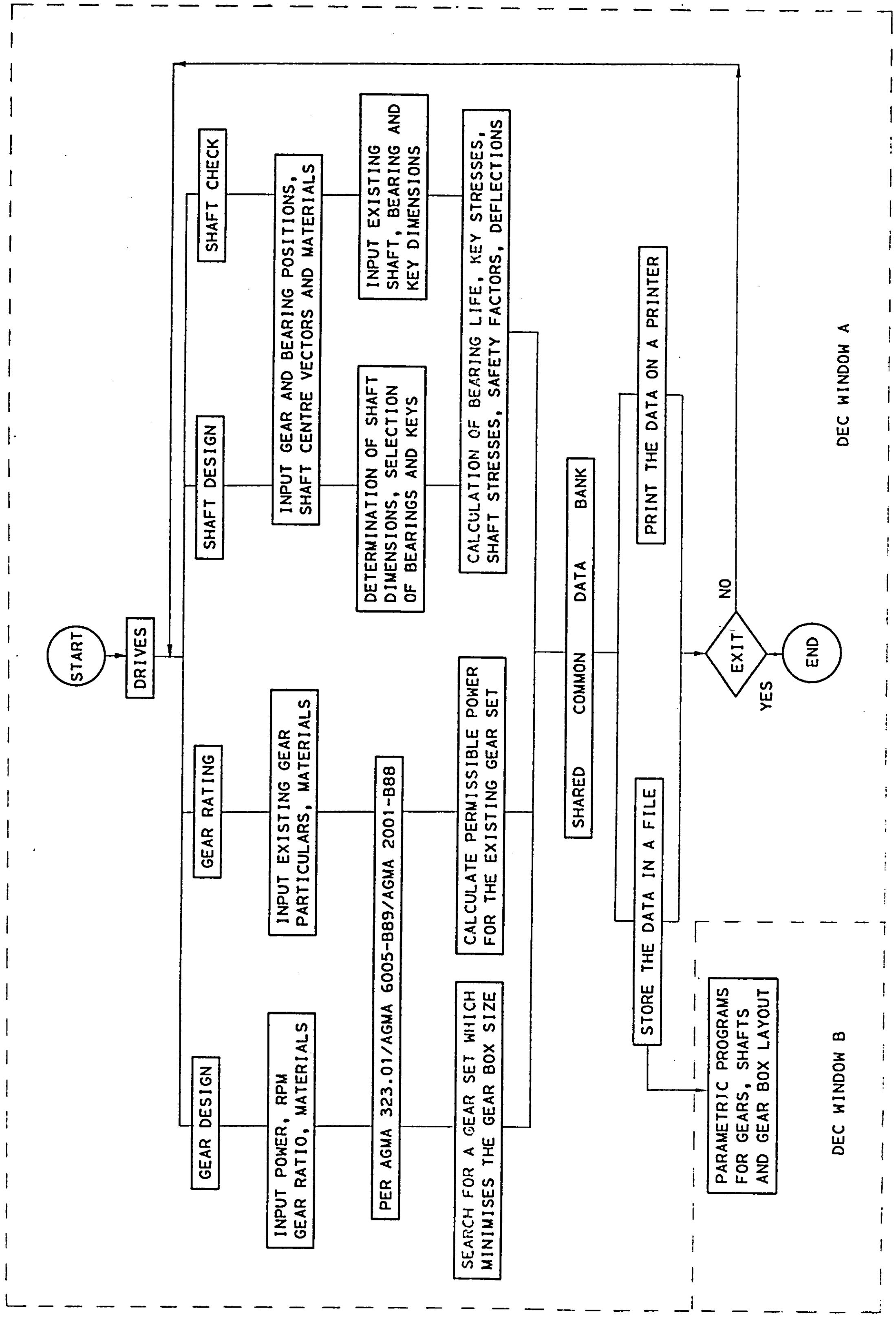


Fig.6-1 Functional Block Diagram

The conventions used in the following diagrams are:

Parallel structure:

```
|
|----MODULE_NAME_1
|
|----MODULE_NAME_2
|
```

meaning one of the modules is selected to be called;

Sequential structure:

```
|
|
-----MODULE_NAME
|
|
```

meaning the module is always called;

Expandable Module:

MODULE_NAME(...)

meaning other modules are called by the named module.

```
DRIVES                              master program of the package
|
-----HLPMSG                         display help message
|
-----INIT_DATA(...)                 initialise data
|
|----DATA_MAN                       data management
|     |
|     |----INIT_DATA(...)           initialise data
|     |
|     |----SAVE_USER(...)           save user data file
|
|----ACTIVITY                       desired action
      |
      |----SAVE_USER(...)           save user data file
      |
      |----GEAR_DESIGN(...)         main module: gear design
      |
      |----SHAFT_DESIGN(...)        main module: shaft design
      |
      |----GEAR_RATING(...)         main module: gear rating
      |
      |----SHAFT_CHECK(...)         main module: shaft check
```

```
GEAR_DESIGN
|
-----GET_STD_CHK                standard chosen check
|
-----APPG_INPUT_CHK             application data check(gear)
|
-----REDN_INPUT_CHK(...)        reduction data check
|
|----SAVE_USER(...)             save user data file
|
|----GET_STD                    choose a gear standard
|
|----APPG_INPUT(...)            application data input(gear)
|
|----REDN_INPUT(...)            reduction data input
|
|----OPT_INPUT(...)             optional data input
|
|----DESIGN_GEAR(...)           gear design calculation
|
|----GEARD_RPT(...)             gear design result report


SHAFT_DESIGN
|
-----RPM_HH_VCT                 init: direction of rotation,
|                               hand of helix, centre vector
-----APPS_INPUT_CHK             application data check(shaft)
|
-----GEAR_INPUT_CHK(...)        gear data check
|
-----INIT_SHF                   initialisation of shafts
|    |
|    -----SHF_INIT(...)         initialisation of each shaft
|
-----SHF_INPUT_CHK(...)         shaft data check
|
-----SHF_AXL_CHK(...)           shaft axial position check
|
-----SHF_KODE                   shaft calculation history
|
|----SAVE_USER(...)             save user data file
|
|----APPS_INPUT(...)            application data input(shaft)
|
|----GEAR_INPUT(...)            existing gear data input
|
|----SHF_INPUT(...)             shaft data input
|
|----DESIGN_SHAFT(...)          shaft design calculation
|
|----SHFD_RPT(...)              shaft design result report
```

```
GEAR_RATING
|
-----GET_STD_CHK                standard chosen check
|
-----APPG_INPUT_CHK             application data check(gear)
|
-----GEAR_INPUT_CHK(...)        gear data check
|
|----SAVE_USER(...)             save user data file
|
|----GET_STD                    choose a gear standard
|
|----APPG_INPUT(...)            application data input(gear)
|
|----GEAR_INPUT(...)            existing gear data input
|
|----OPT_INPUT(...)             optional data input
|
|----RATE_GEAR(...)             gear rating calculation
|
|----GEARC_RPT(...)             gear rating result report


SHAFT_CHECK
|
-----RPM_HH_VCT                 init: direction of rotation,
|                               hand of helix, centre vector
-----APPS_INPUT_CHK             application data check(shaft)
|
-----GEAR_INPUT_CHK(...)        gear data check
|
-----INIT_SHF                   initialisation of shafts
|    |
|    -----SHFT_INIT(...)        initialisation of each shaft
|
-----SHF_INPUT_CHK(...)         shaft data check
|
-----SHF_KODE                   shaft calculation history
|
|----SAVE_USER(...)             save user data file
|
|----APPS_INPUT(...)            application data input(shaft)
|
|----GEAR_INPUT(...)            existing gear data input
|
|----SHF_INPUT(...)             shaft data input
|
|----CHECK_SHAFT(...)           shaft check calculation
|
|----SHFC_RPT(...)              shaft check result report
```

6.2 *Data Organisation and Data Flow*

The DRIVES package is designed to handle gear boxes of up to a maximum of 4 reduction stages. A large number of variables are needed to define the gear boxes. The variables with their values are referred to as data here. The data defining such boxes are divided into three groups, namely, application data, reduction data and shaft data. This breakdown of data is relevant to the scope of validity of each piece of data. The application data are box level data which are valid over all the reductions and shafts, such as transmitted power and modules available for use. Reduction data are related to gears and are valid for each reduction, such as centre distance and power capacity rating of a reduction. The shaft data are related to shafts and other components on shafts. The shaft data are valid for each shaft. Shaft diameters and rated bearing lives are typical shaft data. To facilitate data communication between the four functional branches, all the data in the above three groups are stored in named COMMON blocks of Fortran language, which are accessible to any program module when the named COMMON blocks needed are made known to that module. To avoid confusion of data, unique symbolic names are used throughout all modules for variables defined in the named COMMON blocks.

The data communication between the calculation programs written in Fortran and the parametric programs written in CPROC(which will be discussed in Chapter 7) is realised through data file. The data file is organised according to a model which is common to both the data file and the named

COMMON blocks. This model of data organisation is given in Appendix E. Each line in the data file has a line number and two pieces of data. For instance, line 12 stores the transmitted power(in hp) and the power unit specified by user(hp or kw). These lines of data are organised as follows:

. Lines 001 to 100 are for the Application Data;
. Lines 101 to 200 are for the first Reduction Data;
. Lines 201 to 300 are for the second Reduction Data;
. Lines 301 to 400 are for the third Reduction Data;
. Lines 401 to 500 are for the fourth Reduction Data;
. Lines 501 to 700 are for the first Shaft Data;
. Lines 701 to 900 are for the second Shaft Data;
. Lines 901 to 1100 are for the third Shaft Data;
. Lines 1101 to 1300 are for the fourth Shaft Data;
. Lines 1301 to 1500 are for the fifth Shaft Data.

The number of shafts is the number of reductions plus 1, since one shaft is needed to support the final reduction gear, in addition to shafts with pinions for each reduction (and possibly gears for preceding reductions).

Each subtitle in the data model has a corresponding COMMON block name. The data lines under the subtitle contains the variables in the COMMON block. The COMMON block names and the corresponding subtitles in the data model are listed below:

COMMON block name	Subtitles in the data model
. Application Data:	
CODES	Box level codes
BOX_SPEC	Box specification
GEAR_SPEC	Gear specification
SHAFT_SPEC	Shaft specification
BND_VAL	Design boundary values
STD_DATA	Standard data
MACHINING	Machining data
PRECISION	Precision data
KT_KEYS	Keyway stress concentration
. Reduction Data:	
RDN_CODE	Reduction level codes
TRNSM	Transmission condition
GEARP	Gear particulars
GEARSIZE	Gear physical size
CVECTOR	Gear wheel centre vector
MATERIAL	Material data
MANUFACT	Manufacturing data
RATINGS	Power and stress ratings
FACTORS	Rated factors
FACTORS0	Rating factors
FACTORS1	Pseudo-fixed factors
WHL_DAT	Fabricated wheel data
WHL_WT	Fabricated wheel weight
. Shaft Data:	
SHF_CODE	Shaft level code
SHF_NUM	Shaft definition numbers
SHF_DAT	Shaft design data
GEAR_PP	Gear pitch points
GEAR_LOAD	Gear loads
GEAR_PW	Gear powers
EXT_TQ	External torques
EXT_LOAD	External loads
JOURNALS	Shaft journals + Journal radii + Stressed section positions
SHF_WT	Shaft weight and GD2
KEY_DAT	Key data
BRG_SPEC	Bearing specification
BRG_DIM	Bearing dimension
BRG_POS	Bearing position
BRG_LOAD	Bearing loads
BRG_RATING	Bearing capacity
SHF_KEY	Material allowables
UTL_RATIO	Utilisation ratios
SHF_DFLXN	Deflections
SHF_STRS	Key and section stresses
KT_STRS	Stress concentration factors

Detailed definition of COMMON blocks is given in Appendix F.

The run time interactive data communication is as shown in Fig.6-2. After invocation of the DRIVES package, variables in the COMMON blocks are first initialised to default values from standard data files for boundary values, standard tooth systems, modules available for use, etc., which are based on the practice at the collaborating establishment. The user is then given the opportunity to retrieve his/her own data file saved from a previous run of the package. The values from the user data file, if retrieved, will overwrite the values from the standard data files. The user can now use the keyboard to change the values of any variables in the COMMON blocks, with the existing values shown as default values by the package.

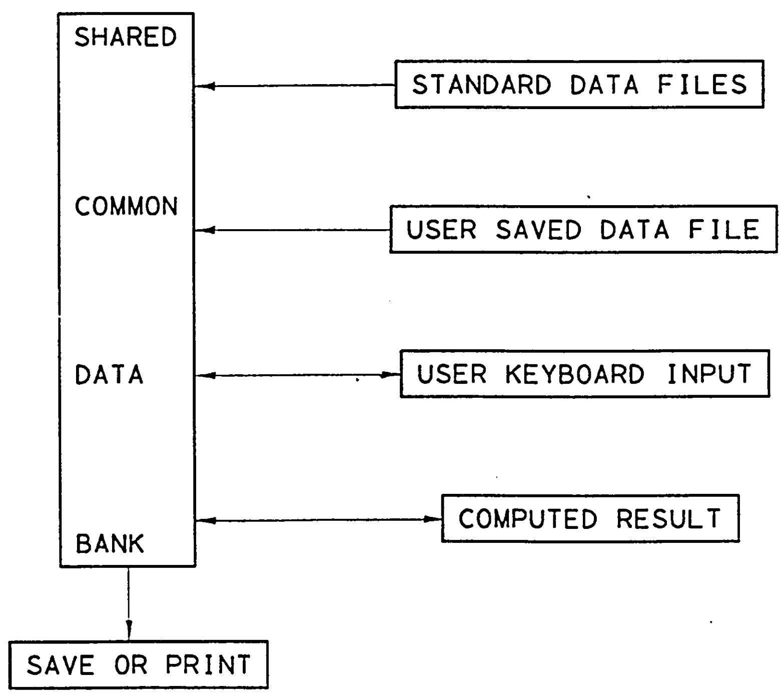


Fig.6-2 Data Flow Diagram

Once the user input has been finished, the computation process can be started. The computing modules get the necessary initial data from the COMMON blocks and put the resulting data from the computations back to the COMMON blocks. When the computations have been finished, the data in the COMMON blocks can be printed or saved to a user data file for use by the parametric programs or for future runs of the DRIVES package.

6.3 *User Interface*

The DRIVES package has a menu-driven user interface. For the input of data, there are facilities for default value prompting, input value validity check, data file retrieval, and easy modification of existing input values. For the output of data, there are facilities for data file saving, and result reporting, either to the displaying screen or a report file which can then be printed. A facility for displaying help messages has also been incorporated into the user interface. The menu system follows closely the main routes of the program calling structure and is given in Appendix G.

7.1 *The CAD System and Parametric Programming*

When the calculations of gears and shafts have been done by running the DRIVES package, the data file saved contains sufficient information(see Appendix E) to create a layout drawing of the gear box. The CAD system used at the collaborating establishment has a graphics language which can be used to write parametric programs to automatically create layout drawings.

The CAD system is a cluster of networked DEC Workstations, running Applicon *EDITOR* [42] on VAX/VMS [25] operating system with multiple DECwindows. Applicon *EDITOR* is a 3D wireframe and solid model editor. For drawing creation, the 3D wireframe editor is most often used. The commands within the 3D editor can create new geometry, edit existing geometry, add dimensions to the geometry, and perform other operations such as database management. The commands can be accessed by a menu system, by typing in from the keyboard, or by retrieval from stored procedures of pre-written lines of commands. The Applicon *EDITOR* command procedures are called CPROCs(Command PROCedures). Most geometric commands require the supply of some parameters to define the geometry to be created or edited. The parameters required by the commands can be supplied by constants typed in from the keyboard or by variables of IAGL[43](Interactive Applicon Graphics Language). The statements of IAGL can be incorporated into a CPROC, making IAGL variables available to the *EDITOR* commands in the

CPROC. IAGL is a graphics language that has the following features in addition to the facilities available in normal programming languages:

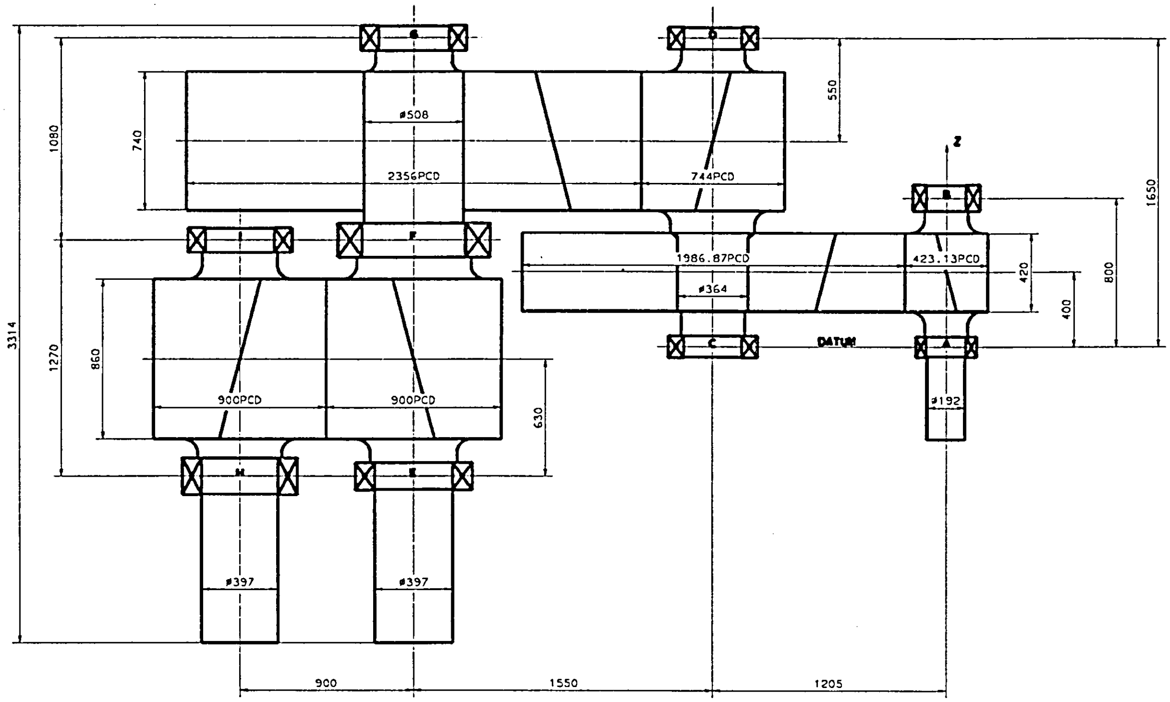
1. geometric variables such as points, vectors, lines, and circles;
2. geometric operations such as vector addition/subtraction and geometry scale up/scale down;
3. geometric functions for such as circle definition, circle centre extraction, and intersecting point calculation;
4. geometric input facilities for getting geometric elements such as points, vectors, lines, and circles;
5. normal programming language facilities such as read/write file statements, if-then-else conditional statements and loop control statements.

Thus by combining IAGL statements and *EDITOR* commands in CPROCs, parametric programs have been written which retrieve data from the data file saved by DRIVES, carry out IAGL geometric calculations on the data retrieved to generate geometric entities, and create and dimension the layout drawing by *EDITOR* commands. Using the multiple DECwindows, it is possible to run the DRIVES package in one window and the parametric programs in another. The simultaneous run of the DRIVES and parametric programs is ideal for design purposes in that the numerical results from the DRIVES can be immediately presented as a layout drawing on the screen. If modification is considered necessary after a layout drawing is created on the screen, the designer can do so by switching to the DRIVES window, making the appropriate modifications to some input data, asking the DRIVES to re-design, and switching back to the parametric window to ask the parametric programs to re-draw the layout drawing.

7.2 *Layout Drawing Creation by Parametric Programs*

To create the layout drawing by parametric programs, the descriptive layout definition used for the DRIVES must be interpreted in the same coordinate system for both the DRIVES and the parametric programs. This coordinate system is specified by the user in the form of positive directions of the X, Y and Z axis as in the front elevation showing the gear centres and pitch circles. X and Y directions can go left, right, up or down, whilst Z direction can go out or into the screen. The right hand rule must be followed. In the layout shown in Fig.5-1a, the X axis goes right, the Y axis goes up and the Z axis goes out of the screen. A data link is thus established between the DRIVES and the parametric programs by the specification of the coordinate system.

The layout drawing of a gear box normally shows at least two views of the gear box, viz., one front elevation showing the arrangement of the shafts and one or more section views and/or an expanded view showing the axial arrangements of components. The expanded view is a section view cutting through all the shaft centres along the centre vectors. If the centre vectors are along one line, expanded view is the same as section view. The A-A section in Fig.7-1 is an expanded view of the box shown. With the coordinate system specified, the centre vectors can be used to calculate the



BEARING DATA:

BEARING	BORE(MM)	O/D(MM)	WIDTH(MM)	LIFE(HOURS)	FIXTURE
A	200.000	310.000	109.000	84709.5	FREE
B	200.000	340.000	140.000	57115.3	FIXED
C	300.000	460.000	118.000	152359.3	FREE
D	300.000	460.000	118.000	69097.8	FIXED
E	400.000	600.000	148.000	566725.8	FREE
F	530.000	780.000	185.000	274450.3	FREE
G	360.000	540.000	134.000	278394.0	FIXED
H	400.000	600.000	200.000	134111.8	FIXED
I	360.000	540.000	134.000	234932.6	FREE

A-A

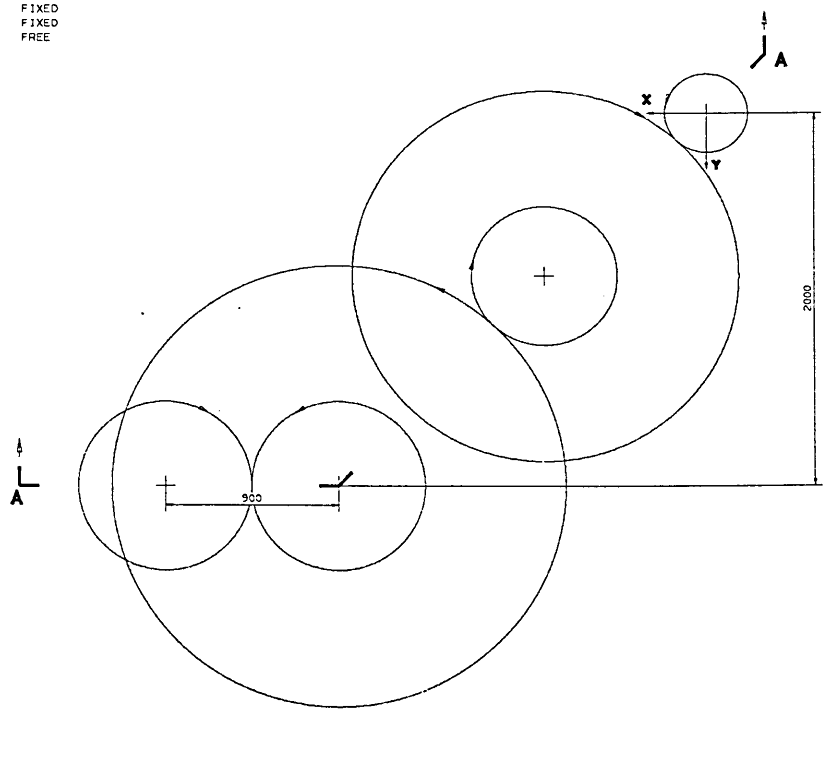


Fig.7-1 The Expanded View of Layout

locations of shaft centres and pitch circles are drawn using these shaft centres on the front elevation.

With the front elevation drawn, a section view can be specified as whether it is a vertical or horizontal section, where it cuts through, and in what direction it is viewed. Parametric programs for individual shaft, bearing, pinion and gear wheel are written to draw the shafts, bearings and gears that are on the specified section line. For an expanded view, the section line cuts through all the shaft centres along the centre vectors. The expanded view can be specified as whether it is a vertical or horizontal expansion and in what directions it is viewed. As the centre vectors may not be along one line, the way how some of the centre vectors are to be rotated to bring them all into one line is defined by specifying the relative position of the input pinion after expansion. For instance, in Fig.7-1, the input pinion after horizontal expansion is on the right. In vertical expansion, the input pinion could equally have been specified to be at the top after expansion. Because the expanded view shows all the shafts, bearings and gears, the main dimensions of the gear box and components are put on the expanded view. The main dimensions shown are:

. centre distance and axial position of each reduction gears

. face width and pitch circle diameters of pinion and gear

. bearing spans, bores, widths and outside diameters

. bearing lives and bearing fixtures

. shaft diameters at keyways and overall axial length

7.3 *Program Structure and Logical Flow*

The parametric programs generating the layout drawings are organised in the same sequence as is outlined in the previous section 7.2. That is, after retrieval of data saved from the DRIVES, a front elevation is created first as the reference drawing for the creation of other section or expanded views. Because of the same reason of limited space as is for Chapter Six, it is only possible to show the program structure of the parametric programs. The IAGL programs are organised into 64 CPROCs and the total number of source program lines is 2067. The master program is named LAYOUT. The two main branches and one sub-branch shown here are FRONT, OTHERS and ELEVATIONS. Full expansion of the program structure can be found in Appendix H. The overall relation between the parametric programs and the DRIVES package is as shown in Fig.6-1, each running in one DECwindow and communicating through the data file. In the following diagrams, the convention is the same as that used in Chapter Six.

```
LAYOUT                  master procedure for layout drawing
|
CHK_VAR(...)            check if variables already declared
|
DCL_ALL                 declare all variables
|
READ_DATA(...)          read data file
|
INIT_DATA               initialise variables with data from file
|
FRONT(...)              create front elevation
|
OTHERS(...)             create other elevations
```

```
FRONT
|
-----FRONT_MODEL(...)       create front model cell
|
-----GET_DIRECTIONS         get directions for the coordinates
|     |
|     -----X_DIRECTION      direction for positive X
|     |
|     -----Y_DIRECTION      direction for positive Y
|     |
|     -----Z_DIRECTION      direction for positive Z
|     |
|     -----CHK_DIRECTIONS   check if right hand rule followed
|           |
|           |----NOTHING    dummy CPROC
|           |
|           |----GET_DIRECTIONS
|
-----CRS_VECTORS            establish centre vectors
|
-----SHF_CENTRES            establish shaft centres
|
-----DRAW_FRONT             draw front elevation
|     |
|     -----WIND_SCALE       determine size of the window
|
-----DIME_FRONT             dimension front elevation
      |
      -----TRI_ARROW        arrow for direction of rotation
OTHERS
|
|----ELEVATIONS(...)        create other elevations
|
|----OPEN_MODEL             list names of cells and open one
|
|----PLOT_WINDOW            plot the drawing shown on screen
|
|----EXIT(...)              shut down and exit

ELEVATIONS
|
-----OPEN_MODEL_ FRONT            open front model cell
|
|----SECTIONS                     create plane sectional view
|     |
|     |----HORIZONTAL_V(...)      horizontal sectional view
|     |
|     |----VERTICAL_V(...)        vertical sectional view
|
|----EXPANDED                     create expanded section view
      |
      |----HORIZONTAL(...)        horizontal expansion
      |
      |----VERTICAL(...)          vertical expansion
```

CHAPTER EIGHT APPLICATIONS OF THE PROGRAM PACKAGE

The development of the DRIVES package has been going on in parallel with the research work on the algorithms for gear design and shaft design. The GEAR DESIGN and GEAR RATING parts of the package were finished by early 1993 and the SHAFT DESIGN and SHAFT CHECK parts at the end of 1993. The whole package, including the parametric programs, was completed by April 1994, with minor modifications since. During the different development stages, the package has been used for different purposes, including real industrial applications. The following sections provide some examples of the application of the package.

8.1 *Analytical Application*

The example problem provided in Chapter Four shows how an optimum design is arrived at and demonstrates the efficiency and effectiveness of the algorithm in finding a global optimum design. To give an indication of how the method developed here compares with others, an example problem from [34] is used:

1. Design specification:

   power to be transmitted P = 100 hp;

   input pinion speed n_p = 1120 r.p.m.;

   gear ratio required m_G = 4, gear ratio tolerance δ = 0%.

2. Gear material allowables:

   pinion and gear are of the same material:

   s_{ac} = 76680 psi and s_{at} = 22450 psi.

3.Factors:

application factors $C_a = K_a = 1$;

life factors $C_L = K_L = 1$;

reliability factors $C_R = K_R = 1$;

load distribution factor $C_m = K_m = 1$.

4.Boundary constraints:

minimum number of pinion teeth $(N_1)_{min} = 18$;

maximum helix angle $\psi_{max} = 35°$;

minimum face contact ratio $(m_F)_{min} = 2$.

5.Gear tooth system:

normal pressure angle $\phi_n = 20°$;

addendum to module ratio $h_a = 1$;

dedendum to module ratio $h_b = 1.35$;

fillet radius to module ratio $r_b = 0.35$.

6.Normal diametral pitches NDP(or normal module) available:

20(1.27), 16(1.5875), 12(2.1167), 10(2.54), 8(3.175),

6(4.2333), 4(6.35), 3(8.4667), 2.5(10.16), 2(12.7)

The objective was to minimise the volume of the gears with the face contact ratio fixed at 2. The optimum design given in [34] was:

NDP	N_1	N_2	ψ	F(mm)	C(mm)	d(mm)	$Fd^2\times10^{-6}$
10	62	248	28.0000	33.99	445.89	178.36	1.0813

where $Fd^2\times10^{-6}$ is the volume index. The dynamic factor $C_v = K_v$ used in [34] was in effect an AGMA quality number between 9 and 10 when equation (1-4-13) for C_v and K_v is used.

In order that a simple comparison can be made between the results, the pinion aspect ratio is set to 0.19 as in the above design. The objective is to minimise the centre distance with a fixed aspect ratio. The minimum centre distance designs for AGMA quality numbers 9 and 10 are found to be, respectively,

NDP	N_1	N_2	ψ	F(mm)	C(mm)	d(mm)	$Fd^2 \times 10^{-6}$
10	63	252	27.9799	35.00	453.00	181.20	1.1492
10	61	244	28.3165	34.00	440.00	176.00	1.0532

Considering the difference in quality numbers used, the minimum centre distance designs are virtually the same as the minimum volume design of [34], which is not unexpected since the pinion aspect ratio is made the same.

A more careful study of the problem and the optimum designs shows that the gear wheel is thin(which makes double helical gears impractical) and the axial load generated by the large helix angle may cause instability in the thin gear wheel. This is a result of the volume minimisation without control over the gear blank proportion. Only face contact ratio was used in [34] to control the face width, which is not sufficient. The pinion aspect ratio of 0.19 of the optimum design means that the pinion pitch circle diameter is about 5 times the face width, and the gear pitch circle diameter is about 20 times the face width for a gear ratio of 4. A more reasonable gear blank proportion can be achieved if an aspect ratio of 0.5 is used, which gives a gear pitch circle diameter to face width ratio of 8. To reduce the axial load generated by the helix action, helix angle is normally kept below 20 degrees for single helical gears. Thus by changing the pinion aspect ratio to 0.5 and the maximum helix angle to 20 degrees, more practical optimum designs can be found for the modified problem. The new minimum centre distance designs are, for AGMA quality numbers 9 and 10, respectively:

NDP	N_1	N_2	ψ	F(mm)	C(mm)	d(mm)	$Fd^2 \times 10^{-6}$
10	45	180	15.8204	59.00	297.00	118.80	0.8327
10	44	176	16.2321	58.00	291.00	116.40	0.7858

These two designs actually have smaller volume indexes $Fd^2 \times 10^{-6}$ than the minimum volume design in [34]. The initial search intervals for centre distances are [294,318] for quality grade 9 and [288,312] for quality grade 10. A similar exercise to that in Chapter Four has been carried out to exhaust all possible combinations at every centre distance and plot the maximum power combinations for a range of centre distances, which is shown in Fig.8-1. It can be seen from the plots that the minimum centre distances found by the algorithm are global minimum.

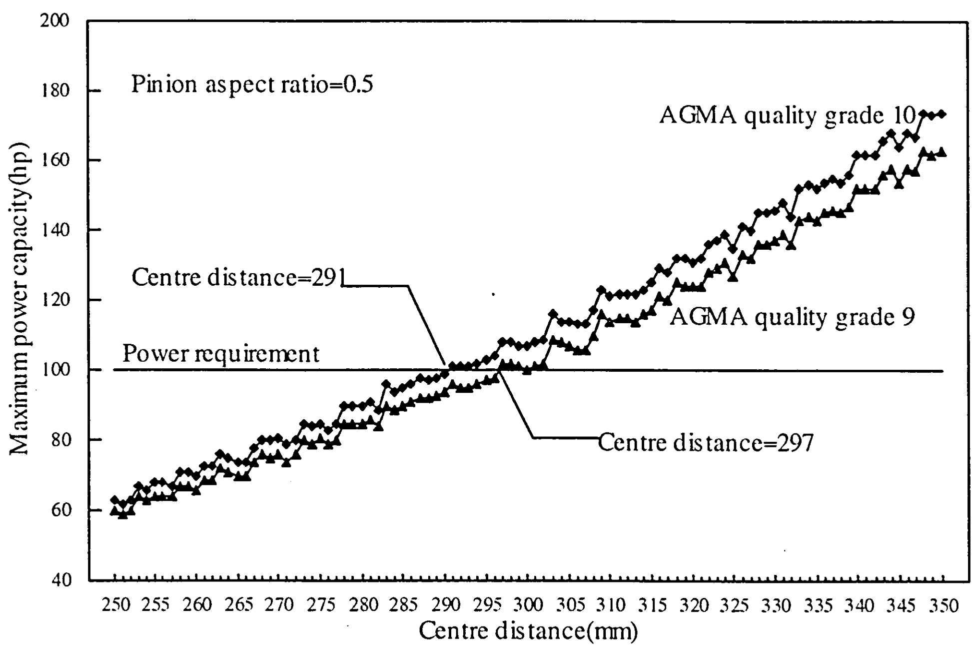


Fig.8-1 Maximum power combinations

The numbers of capacity evaluations for the original problem are 65 and 61 for AGMA quality grades 9 and 10, respectively.

For the modified problem, they are reduced significantly to 15 and 11 for quality grades 9 and 10, respectively, because of the smaller range of helix angle used.

8.2 *Industrial Application*

During the different stages of development of the DRIVES package, it has been tested for various gear design requirements and numerous real engineering design problems have been solved. The package has been used at the collaborating establishment for both tender and contract purposes.

8.2.1 *Application Example 1*

A recent example is the use of the package for the design of a double reduction gear box for the rotary crop shear drive of a steckel mill. The first enquiry specification was received by the collaborating establishment in March 1993 and the gear box is now being assembled. The overall arrangement of the crop shear and the gear box is shown in Fig.8-2. The output shaft of the gear box drives the top drum of the shear through the drive spindle. The material flow is into the paper. There is a blade on the top and bottom drum of the shear as shown in Fig.8-3. The shear is activated intermittently for the cutting action. This intermittent cutting action introduced two special features about a rotary crop shear drive. Firstly, the transmitted power for the gear box is worked out from the cutting torque at the output shaft, instead of the nominal motor power which is much smaller. The cutting torque is not only powered by the motor,

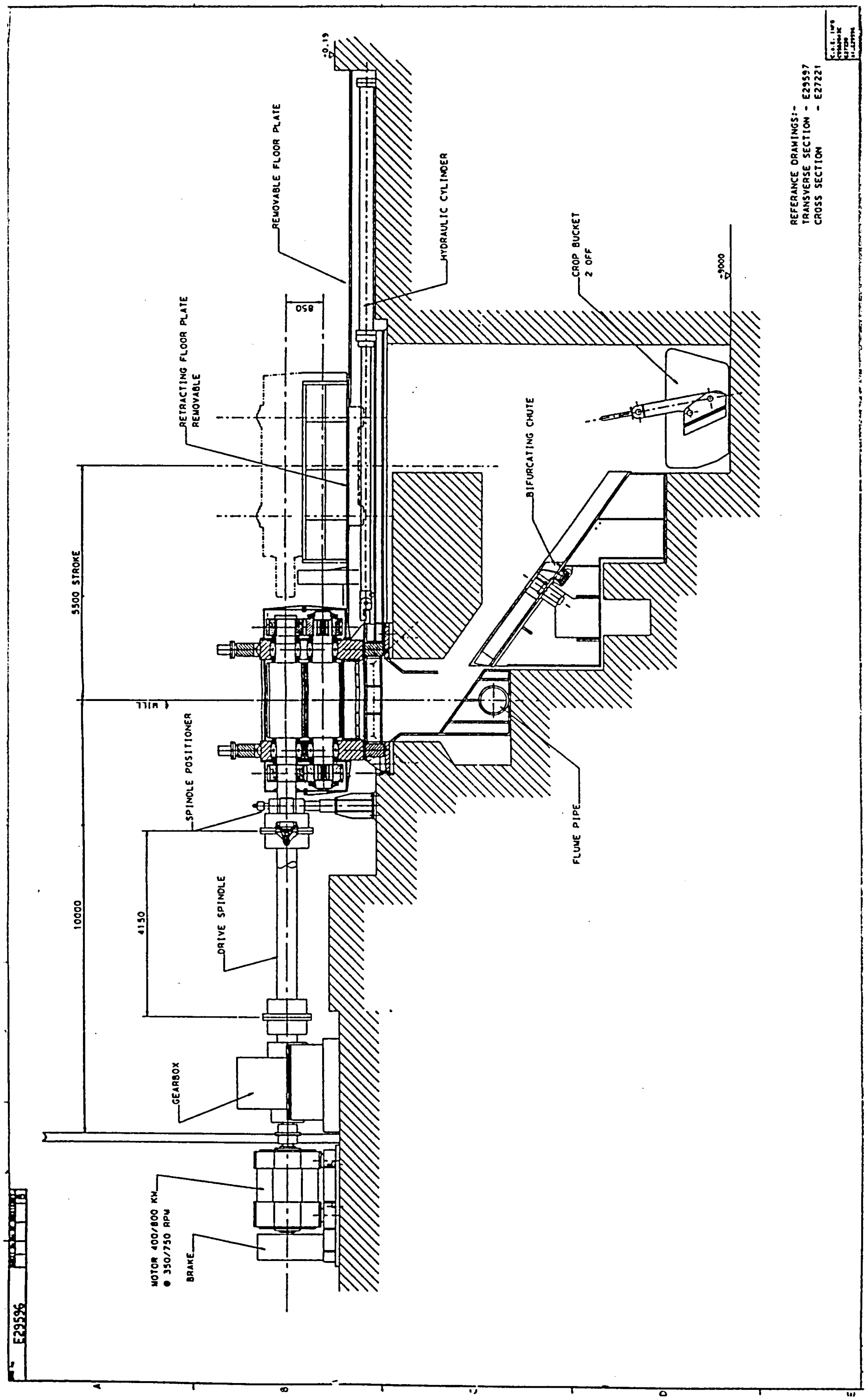


Fig.8-2 Overall Arrangement of Crop Shear and Gear Box

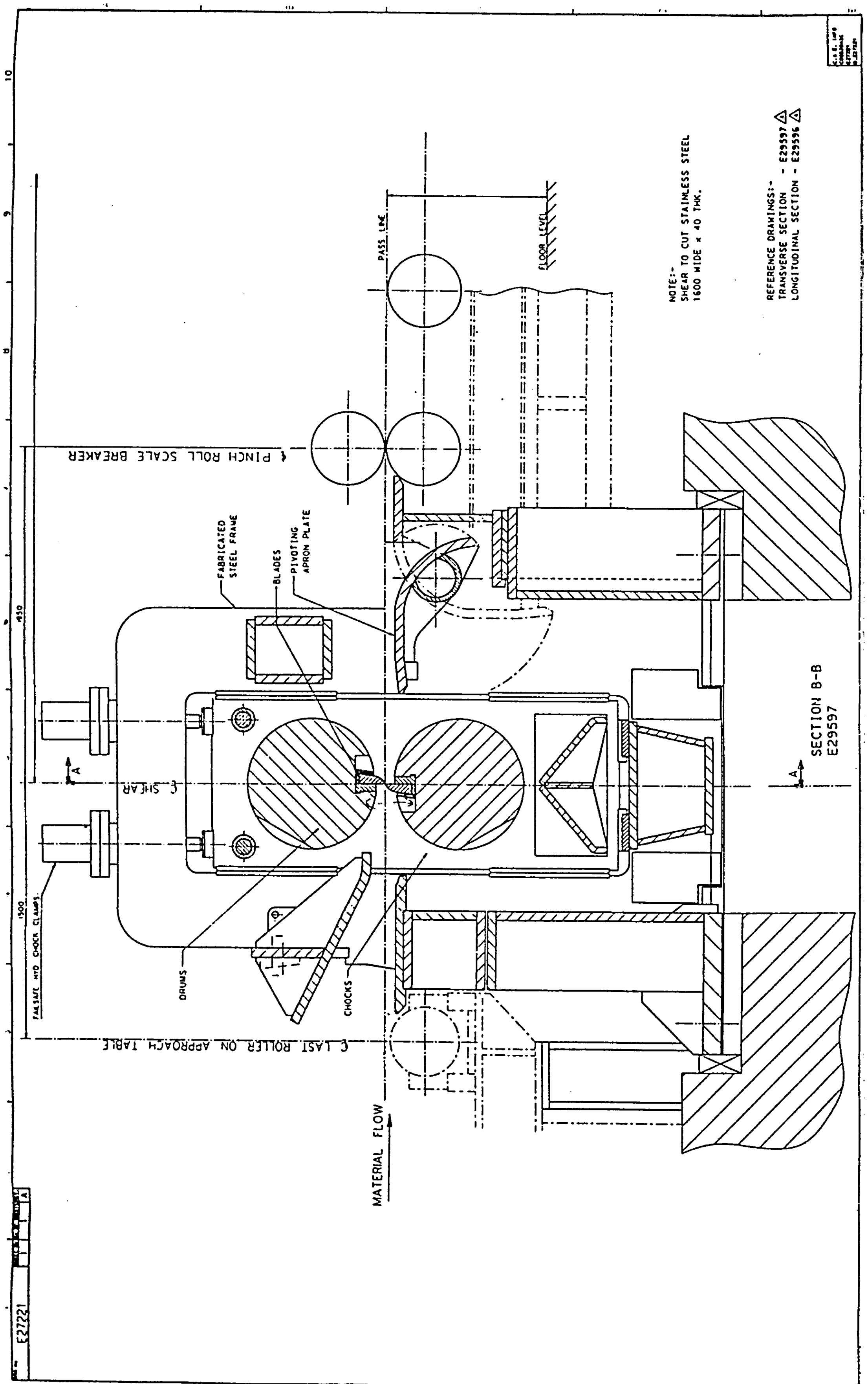


Fig.8-3 Cross Section of Rotary Crop Shear

but also by the stored energy in the drive inertia system built up prior to the cutting action. Thus the application factors C_a and K_a can have values of unity or near unity. Secondly, the action of cutting happens in a very short time and the cutting torque is seen only by a few teeth on the output shaft gear. To avoid the same teeth of gears in the gear train taking the cutting torque, it is desirable for the gear sets to have a hunting ratio which means each and every tooth of the pinion will, in its turn, mesh with any particular tooth of the gear. Hunting ratio is achieved when there are no common factors between the number of pinion teeth and number of gear teeth.

Summarising the gear box specification, the problem is to find the minimum centre distance design which satisfies the following requirement:

1.Design specification:

power to be transmitted P = 11428 kw, for

cutting torque of 1.76 MN at 62.01 r.p.m. at output shaft;

input pinion speed n_p = 700 r.p.m., non-reversing;

overall gear ratio required m_o = 11.288,

gear ratio tolerance δ = 0.5%;

gear box is to be of double reduction box;

gears are to be of single helical type.

2.Gear materials:

pinion carburised and case hardened to HRC 58:

s_{ac1} = 203000 psi, s_{at1} = 56000 psi;

gear through hardened to BHN 320:

s_{ac2} = 133770 psi, s_{at2} = 40480 psi.

3. Factors:

application factor for pitting resistance $C_a = 1.00$,

application factor for bending strength $K_a = 1.15$;

life factors $C_L = K_L = 1$;

reliability factors $C_R = K_R = 1$.

4. Boundary constraints:

minimum number of pinion teeth $(N_1)_{min} = 18$;

maximum helix angle $\psi_{max} = 15°$,

minimum helix angle $\psi_{min} = 8°$;

minimum overlap ratio $(m_F)_{min} = 1.1$.

5. Gear tooth system:

normal pressure angle $\phi_n = 20°$,

addendum to module ratio $h_a = 1$,

dedendum to module ratio $h_b = 1.4$,

fillet radius to module ratio $r_b = 0.4$.

6. Other given design requirements:

pinion aspect ratio $\lambda = 1$;

AGMA gear accuracy grade $Q_v = 11$;

prime number larger than 100 cannot be used for the gear;

normal modules m_n available for use(in mm):

4,5,6,7,8,9,10,12,14,16,18,20,22,25,30,34,40.

The overall gear ratio of 11.288 is split into 4.11 and 2.75 for the first and second reductions, respectively, using the following empirical formulae from the collaborating establishment:

$$m_{G1} = \left(\frac{3}{2} m_O\right)^{1/2} \qquad (8\text{-}2\text{-}1)$$

$$m_{G2} = \left(\frac{2}{3} m_O\right)^{1/2} \qquad (8\text{-}2\text{-}2)$$

where

m_O = overall gear ratio for the box

m_{G1} = gear ratio for the first reduction

m_{G2} = gear ratio for the second reduction

The gear particulars of the gear box as built are as follows

GEAR DESIGN MAIN RESULT REPORT FOR 1ST REDUCTION

```
RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:-------------------------------------------------------------
CENTRE DISTANCE= 1185.000 MM            FACEWIDTH= 440.00 MM
NORMAL MODULE=20.0 MM                   HELIX ANGLE= 9  7 34( 9.1261, SINGLE)
NO. OF PINION TEETH= 23( 23 HUNTING)    NO. OF WHEEL TEETH=  94
PINION P.C.D.= 465.897 MM               WHEEL P.C.D.= 1904.103 MM
PINION ADDN. MODIF. COEF.= 0.21673      WHEEL ADDN. MODIF. COEF.=-0.21673
---RATIOS:-----------------------------------------------------------------
ACTUAL GEAR RATIO= 4.09(ERROR= -0.56%)  REQUIRED GEAR RATIO= 4.11
OVERLAP(FACE CONTACT) RATIO= 1.11       TRANSVERSE CONTACT RATIO=1.64
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):-------------------------------
MESH POWER REQUIREMENT=11428.0 KW       PINION RPM= 700.00
PINION PITTING RESISTANCE=25813.2 KW    SERVICE FACTOR= 2.26( 1.00 REQUIRED)
WHEEL PITTING RESISTANCE=12788.4 KW     SERVICE FACTOR= 1.12( 1.00 REQUIRED)
PINION BENDING STRENGTH=16402.6 KW      SERVICE FACTOR= 1.65( 1.15 REQUIRED)
WHEEL BENDING STRENGTH=12034.7 KW       SERVICE FACTOR= 1.21( 1.15 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:-----------------------------
PINION HARDNESS= 58.0HRC                WHEEL HARDNESS=320.0BHN
PINION CONTACT ALLOWABLE=203000.0 psi   WHEEL CONTACT ALLOWABLE=133770.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi   WHEEL BENDING(UNIDIREC)= 40479.0 psi
MANUFACTURING METHOD=FINISH GROUND      AGMA QUALITY GRADE=11
===========================================================================
```

GEAR DESIGN MAIN RESULT REPORT FOR 2ND REDUCTION

```
RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:-------------------------------------------------------------
CENTRE DISTANCE= 1435.000 MM            FACEWIDTH= 800.00 MM
NORMAL MODULE=30.0 MM                   HELIX ANGLE=10 42 38(10.7106, SINGLE)
NO. OF PINION TEETH= 25( 25 HUNTING)    NO. OF WHEEL TEETH=  69
PINION P.C.D.= 763.298 MM               WHEEL P.C.D.= 2106.702 MM
PINION ADDN. MODIF. COEF.= 0.15784      WHEEL ADDN. MODIF. COEF.=-0.15783
---RATIOS:-----------------------------------------------------------------
ACTUAL GEAR RATIO= 2.76(ERROR= -0.1%)   REQUIRED GEAR RATIO= 2.76
OVERLAP(FACE CONTACT) RATIO= 1.58       TRANSVERSE CONTACT RATIO=1.64
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):-------------------------------
MESH POWER REQUIREMENT=11428.0 KW       PINION RPM= 171.28
PINION PITTING RESISTANCE=23960.5 KW    SERVICE FACTOR= 2.10( 1.00 REQUIRED)
WHEEL PITTING RESISTANCE=11870.6 KW     SERVICE FACTOR= 1.04( 1.00 REQUIRED)
PINION BENDING STRENGTH=15916.5 KW      SERVICE FACTOR= 1.60( 1.15 REQUIRED)
WHEEL BENDING STRENGTH=11661.5 KW       SERVICE FACTOR= 1.17( 1.15 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:-----------------------------
PINION HARDNESS= 58.0HRC                WHEEL HARDNESS=320.0BHN
PINION CONTACT ALLOWABLE=203000.0 psi   WHEEL CONTACT ALLOWABLE=133770.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi   WHEEL BENDING(UNIDIREC)= 40479.0 psi
MANUFACTURING METHOD=FINISH GROUND      AGMA QUALITY GRADE=11
===========================================================================
```

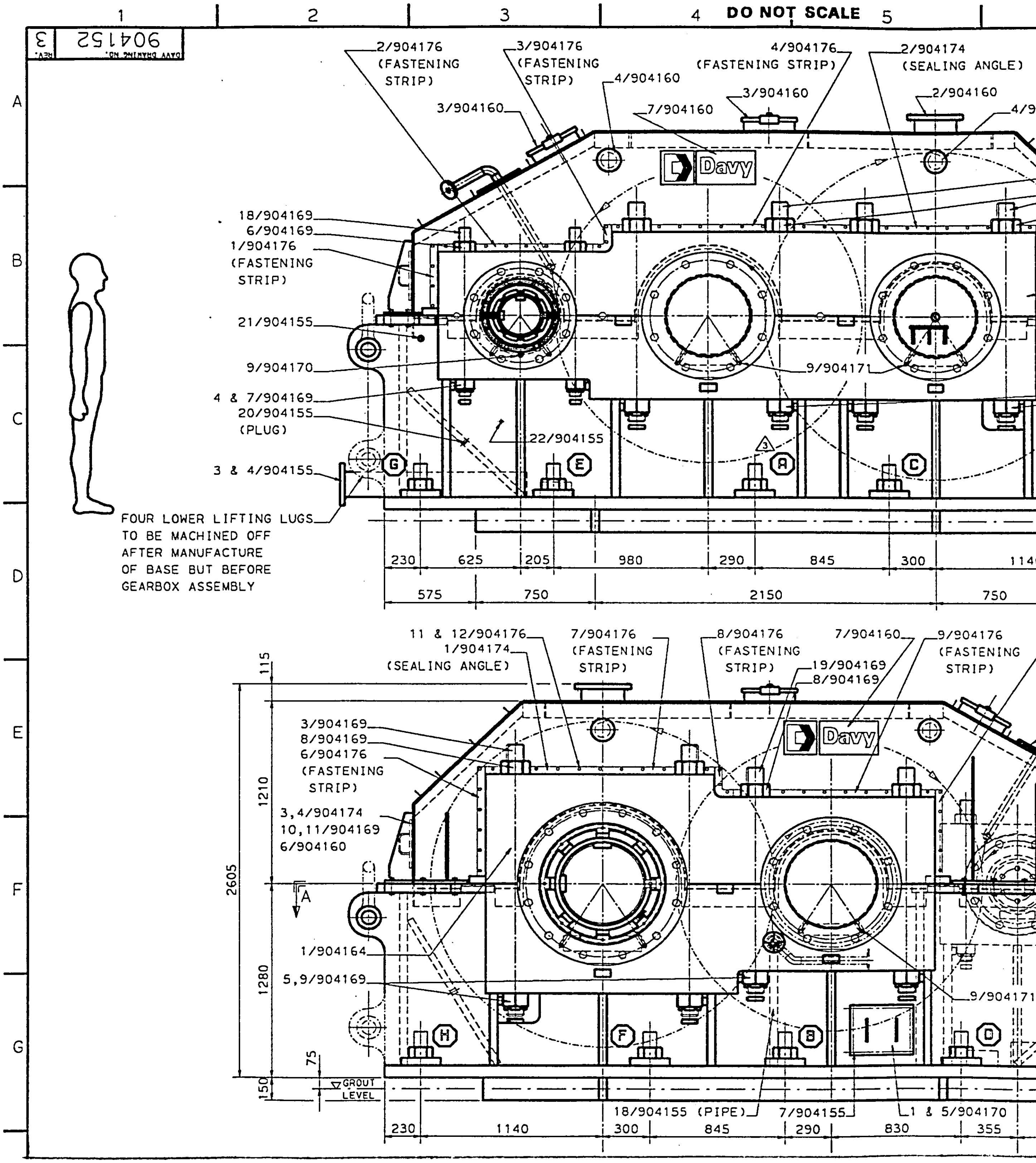


Fig.8-4a Gear Box for Rotary Crop Shear

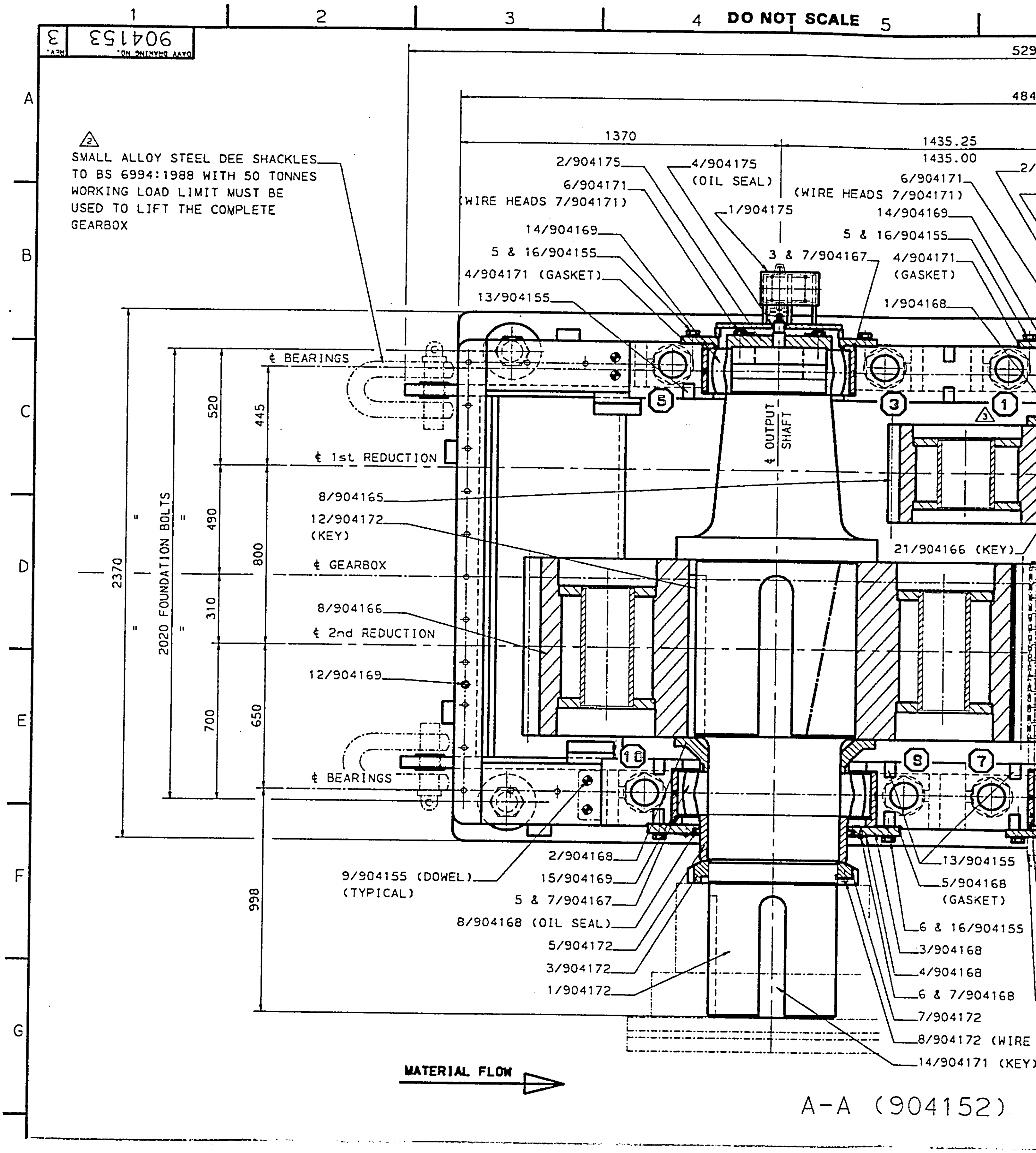
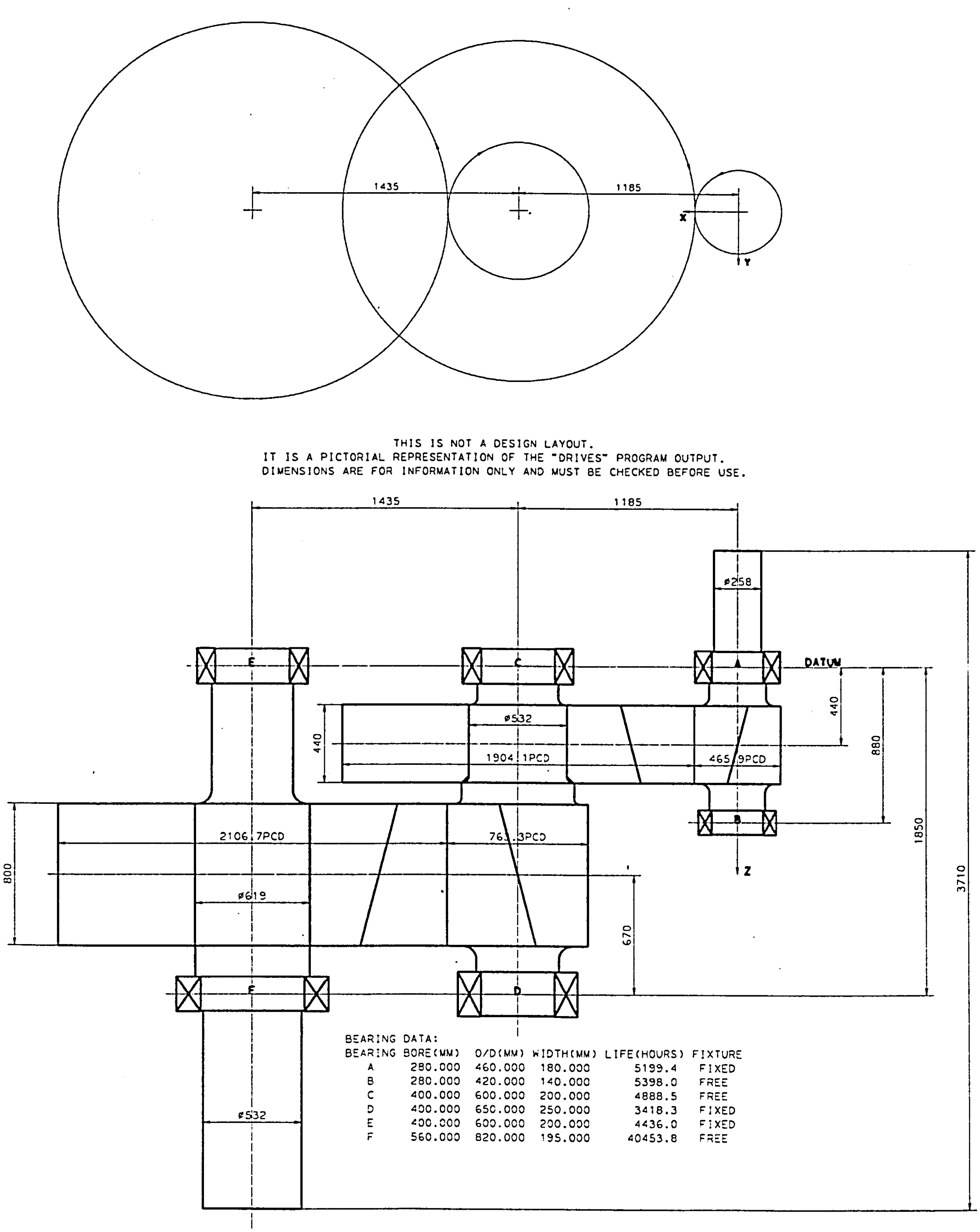


Fig.8-4b Gear Box for Rotary Crop Shear



BEARING	BORE(MM)	O/D(MM)	WIDTH(MM)	LIFE(HOURS)	FIXTURE
A	280.000	460.000	180.000	5199.4	FIXED
B	280.000	420.000	140.000	5398.0	FREE
C	400.000	600.000	200.000	4888.5	FREE
D	400.000	650.000	250.000	3418.3	FIXED
E	400.000	600.000	200.000	4436.0	FIXED
F	560.000	820.000	195.000	40453.8	FREE

Fig.8-5 Layout Drawn by the Parametric Programs

Fig.8-4a and Fig.8-4b are the general arrangement drawing of the gear box. At the time of designing the gear box, only the GEAR DESIGN part of the DRIVES package had been finished and used for the design of the box. The design of shaft components was done by a combination of long hand calculation and using individual programs for reaction calculation and bearing selection which were yet to be incorporated in the DRIVES package. The DRIVES package has been run since for comparison of the designs of the shaft components. The layout has been drawn by the parametric programs for comparison as well, as shown in Fig.8-5. The bearings selected by the DRIVES are the same as in the 'as built' design and the shaft diameters at keyways are virtually the same. However, it took the design engineer about four days to do the shaft component design by long hand calculation as against less than two hours by using the DRIVES.

The service factors in the above results for the 'as built' design are defined as follows:

$$C_{SF} = C_a\left(\frac{C_R}{C_L}\right)^2 \qquad (8\text{-}2\text{-}3)$$

$$K_{SF} = K_a\left(\frac{K_R}{K_L}\right)^2 \qquad (8\text{-}2\text{-}4)$$

where

C_{SF} = service factor for pitting resistance

K_{SF} = service factor for bending strength

Since life factors C_L and K_L, and reliability factors C_R and K_R are set to unity, the service factors C_{SF} and K_{SF} can be compared directly with the application factors C_a and K_a. It can be seen that the gear set for the first reduction is over designed with service factors higher than necessary. This is because the 'as built' design is an evolution of some earlier designs based on different gear box specifications. At the initial tender stage, the cutting torque specified by the customer was 1.89 MN and the overall gear ratio 10.757. This specification was later changed to cutting torque 1.76 MN and overall gear ratio 10.73. The final contract specification was cutting torque 1.76 MN and overall gear ratio 11.288. Had the final specification been given at the initial stage, the optimised design would have been as follows

```
             GEAR DESIGN MAIN RESULT REPORT FOR 1ST REDUCTION

RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:--------------------------------------------------------------
CENTRE DISTANCE= 1128.000 MM            FACEWIDTH= 441.00 MM
NORMAL MODULE=22.0 MM                   HELIX ANGLE= 9 57 36( 9.9599, SINGLE)
NO. OF PINION TEETH= 20( 20 HUNTING)    NO. OF WHEEL TEETH=  81
PINION P.C.D.= 446.733 MM               WHEEL P.C.D.= 1809.267 MM
PINION ADDN. MODIF. COEF.= 0.23655      WHEEL ADDN. MODIF. COEF.=-0.23656
---RATIOS:------------------------------------------------------------------
ACTUAL GEAR RATIO= 4.05(ERROR= -1.6%)   REQUIRED GEAR RATIO= 4.11
OVERLAP(FACE CONTACT) RATIO= 1.10       TRANSVERSE CONTACT RATIO=1.60
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):--------------------------------
MESH POWER REQUIREMENT=11428.0 KW       PINION RPM= 700.00
PINION PITTING RESISTANCE=23822.2 KW    SERVICE FACTOR= 2.08( 1.00 REQUIRED)
WHEEL PITTING RESISTANCE=11802.0 KW     SERVICE FACTOR= 1.03( 1.00 REQUIRED)
PINION BENDING STRENGTH=17069.0 KW      SERVICE FACTOR= 1.72( 1.15 REQUIRED)
WHEEL BENDING STRENGTH=12390.0 KW       SERVICE FACTOR= 1.25( 1.15 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:------------------------------
PINION HARDNESS= 58.0HRC                WHEEL HARDNESS=320.0BHN
PINION CONTACT ALLOWABLE=203000.0 psi   WHEEL CONTACT ALLOWABLE=133770.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi   WHEEL BENDING(UNIDIREC)= 40479.0 psi
MANUFACTURING METHOD=FINISH GROUND      AGMA QUALITY GRADE=11
============================================================================
```

GEAR DESIGN MAIN RESULT REPORT FOR 2ND REDUCTION

```
RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:-------------------------------------------------------------
CENTRE DISTANCE= 1410.000 MM              FACEWIDTH= 745.00 MM
NORMAL MODULE=30.0 MM                     HELIX ANGLE=14 30 51(14.5143, SINGLE)
NO. OF PINION TEETH= 24( 24 HUNTING)      NO. OF WHEEL TEETH=  67
PINION P.C.D.= 743.736 MM                 WHEEL P.C.D.= 2076.264 MM
PINION ADDN. MODIF. COEF.= 0.15813        WHEEL ADDN. MODIF. COEF.=-0.15813
---RATIOS:-----------------------------------------------------------------
ACTUAL GEAR RATIO= 2.79(ERROR=  0.2%)     REQUIRED GEAR RATIO= 2.79
OVERLAP(FACE CONTACT) RATIO= 1.98         TRANSVERSE CONTACT RATIO=1.60
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):-------------------------------
MESH POWER REQUIREMENT=11428.0 KW         PINION RPM= 172.84
PINION PITTING RESISTANCE=23132.5 KW      SERVICE FACTOR= 2.02( 1.00 REQUIRED)
WHEEL PITTING RESISTANCE=11460.3 KW       SERVICE FACTOR= 1.00( 1.00 REQUIRED)
PINION BENDING STRENGTH=16010.5 KW        SERVICE FACTOR= 1.61( 1.15 REQUIRED)
WHEEL BENDING STRENGTH=11722.7 KW         SERVICE FACTOR= 1.18( 1.15 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:-----------------------------
PINION HARDNESS= 58.0HRC                  WHEEL HARDNESS=320.0BHN
PINION CONTACT ALLOWABLE=203000.0 psi     WHEEL CONTACT ALLOWABLE=133770.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi     WHEEL BENDING(UNIDIREC)= 40479.0 psi
MANUFACTURING METHOD=FINISH GROUND        AGMA QUALITY GRADE=11
===========================================================================
```

It is interesting to compare the 'as built' design and the optimised design with a 'would be' past design, had the DRIVES package not been developed. The procedure for gear design prior to the development of DRIVES was to use a centre distance and face width estimating program based on pitting resistance and square pinion. After obtaining the centre distance and face width, a cut and trial method was used to determine the module, number of pinion teeth, number of gear teeth and helix angle, starting from initial values of 22 teeth for pinion and 11 degrees for helix angle(single helical). The centre distance C and face width F given by the estimating program are:

C = 1282 mm and F = 502 mm, for first reduction

C = 1488 mm and F = 794 mm, for second reduction

Assuming exact gear ratios are achieved for the 'would be' past design, the pitch circle diameters of pinion and gear, d and D, for the above centre distances are:

d = 501.76 mm and D = 2062.24 mm, for first reduction

d = 793.60 mm and D = 2182.40 mm, for second reduction

To give a simple indication how the three designs compare, the rotor volume of gears in the gear train is used. The rotor volume V of a gear set is

$$V = \frac{\pi}{4}(d^2 + D^2)F \qquad (8\text{-}2\text{-}5)$$

The rotor volume calculation for each reduction of the three designs can be tabulated as follows

	As Built		Optimised		Would be	
	1st	2nd	1st	2nd	1st	2nd
d	465.90	763.30	446.73	743.74	501.76	793.60
D	1904.10	2106.70	1809.27	2076.26	2062.24	2182.40
F	440.00	800.00	441.00	745.00	502.00	794.00
V	1.3279	3.1547	1.2029	2.8460	1.7180	3.3629 $\times 10^9$

Using the total rotor volume of the 'would be' past design as the basis for volume comparison, the total rotor volumes of the 'as built' design and the 'optimised' design are 88.2% and 79.7%, respectively. This is a saving of 11.8% and 20.3% in volume or weight. Although the rotary volume cannot be considered directly as the cost of a gear box, the comparison is nevertheless a good indication of the relative cost. For a comparison of time, it would have taken an engineer from two to four hours to work out by long hand calculation an acceptable design; whilst it took only about ten minutes using the DRIVES to arrive at an optimum design, most of which was data input time.

8.2.2 *Application Example 2*

Another recent example is the use of the package for the design of a two speed gear box for the main drive of a cold strip mill. The first enquiry specification was received by the collaborating establishment in August 1993 and the gear box was shipped to customer in May 1994. The finished gear box is shown in Fig.8-6a and Fig.8-6b. The gear box is a combination of reduction gear sets and mill pinions. The reduction gears have a low speed set and a high speed set, engaged by a mechanical clutch. The top and bottom mill pinions drive the top and bottom rolls of the mill, each taking theoretically 50% of the total input power. For gear capacity rating and shaft diameter design, it is customary and prudent to use 60% of the total input power. The DRIVES package is capable of dealing with the designs of reduction box, mill pinions alone, or combined gear box of reduction gears and mill pinions which is called compact drive at the collaborating establishment. Similar to the design of the rotary crop shear drive box, because of the development stage the DRIVES was in, only the gears were designed by the GEAR DESIGN part of the DRIVES package; the shaft components were designed manually with the help of some individual programs. Again, the DRIVES package has since been run for comparison.

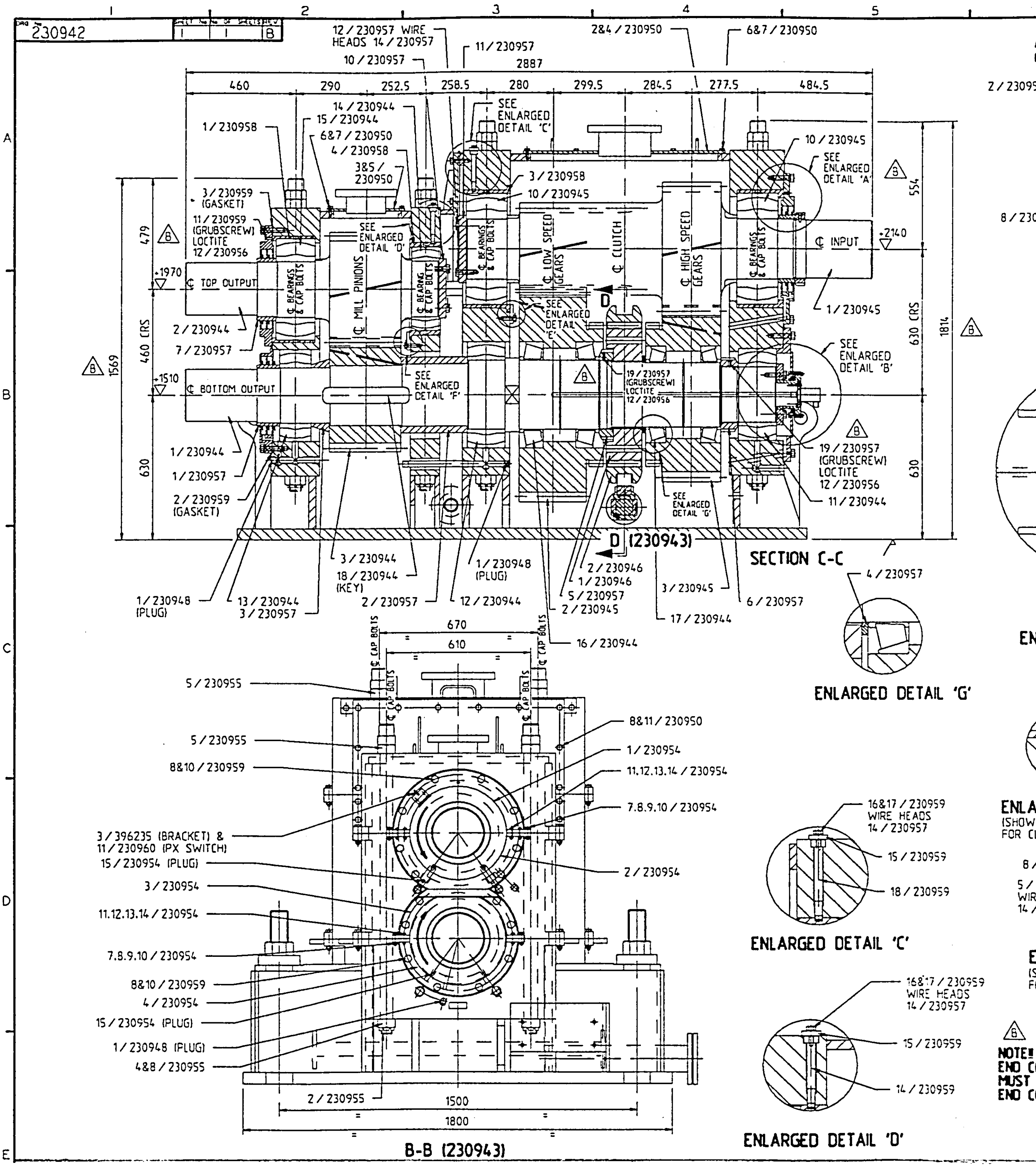


Fig.8-6a Gear Box for Cold Strip Mill

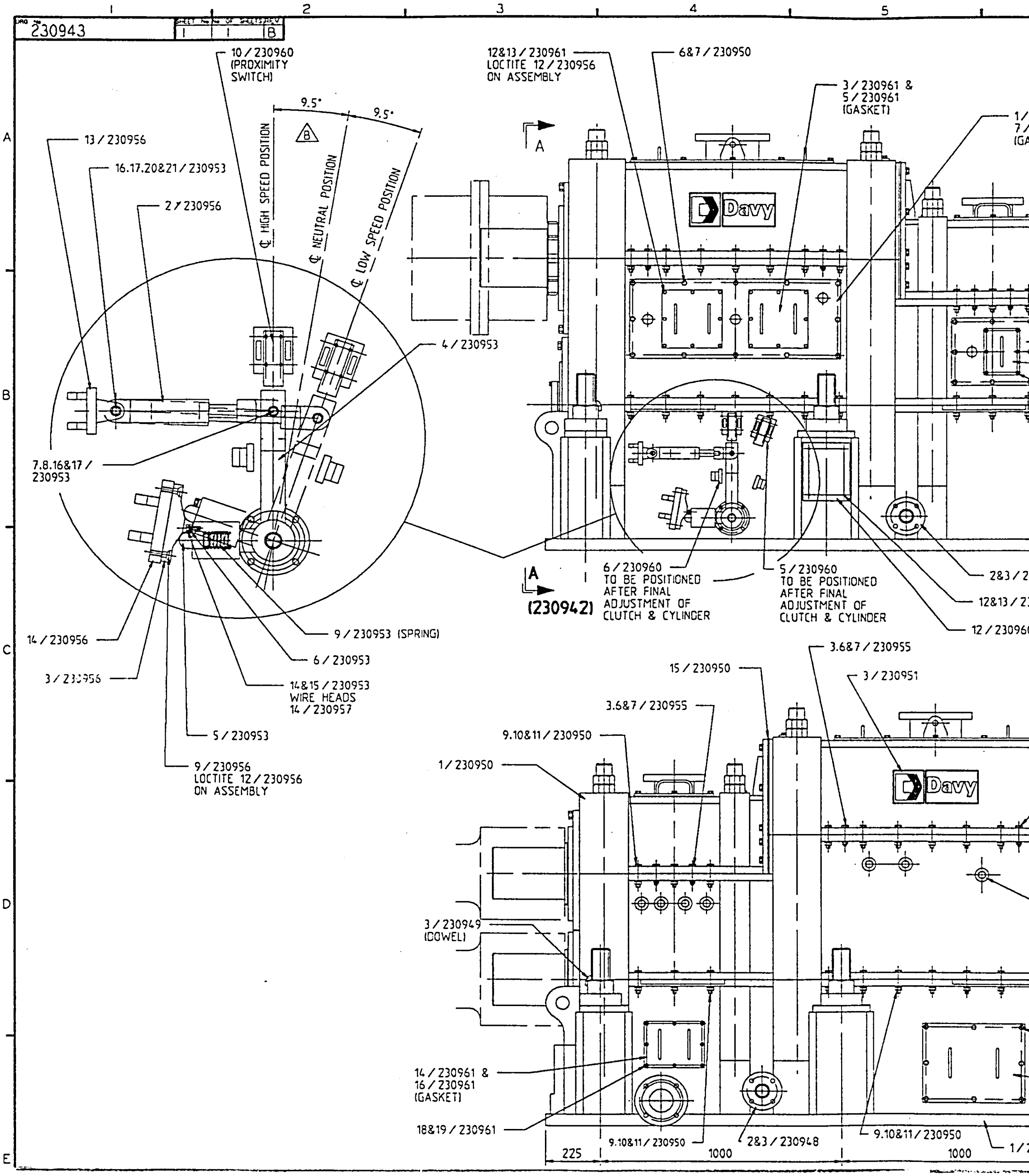


Fig.8-6b Gear Box for Cold Strip Mill

The design requirements can be summarised as follows.

1. Design specification:

power to be transmitted P = 3800 kw

input pinion speed n_p = 570 r.p.m., non-reversing;

low speed gear ratio required m_G = 2.51,

high speed gear ratio required m_G = 1.32,

gear ratio tolerance δ = 2%;

gear box is to be of single reduction compact drive box;

gears are to be of single helical type.

2. Gear materials:

both pinion and gear case hardened to HRC 58:

$s_{ac1} = s_{ac2}$ = 203000 psi, $s_{at1} = s_{at2}$ = 56000 psi;

3. Factors:

application factors $C_a = K_a$ = 1.50,

life factors $C_L = K_L$ = 1;

reliability factors $C_R = K_R$ = 1.

4. Boundary constraints, gear tooth system and other given design requirements are the same as rotary crop shear box.

Since the low speed gears have larger tangential loads, the gear box has been mainly designed for the low speed gears; the high speed gears will always fit into the centre distance determined by the low gears. The minimum centre distance design for the low speed gears(1ST REDUCTION) and mill pinions(2ND REDUCTION) given by the GEAR DESIGN part of the DRIVES package is as follows.

GEAR DESIGN MAIN RESULT REPORT FOR 1ST REDUCTION

```
RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:----------------------------------------------------------
CENTRE DISTANCE=  537.000 MM            FACEWIDTH= 306.00 MM
NORMAL MODULE=16.0 MM                   HELIX ANGLE=10 30 17(10.5046, SINGLE)
NO. OF PINION TEETH= 19( 19 HUNTING)    NO. OF WHEEL TEETH=  47
PINION P.C.D.= 309.182 MM               WHEEL P.C.D.=  764.818 MM
PINION ADDN. MODIF. COEF.= 0.15485      WHEEL ADDN. MODIF. COEF.=-0.15485
---RATIOS:--------------------------------------------------------------
ACTUAL GEAR RATIO= 2.47(ERROR= -1.4%)   REQUIRED GEAR RATIO= 2.51
OVERLAP(FACE CONTACT) RATIO= 1.11       TRANSVERSE CONTACT RATIO=1.58
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):----------------------------
MESH POWER REQUIREMENT= 3800.0 KW       PINION RPM= 570.00
PINION PITTING RESISTANCE= 3954.7 KW    SERVICE FACTOR= 1.56( 1.50 REQUIRED)
WHEEL PITTING RESISTANCE= 3954.7 KW     SERVICE FACTOR= 1.56( 1.50 REQUIRED)
PINION BENDING STRENGTH= 3870.9 KW      SERVICE FACTOR= 1.53( 1.50 REQUIRED)
WHEEL BENDING STRENGTH= 3917.3 KW       SERVICE FACTOR= 1.55( 1.50 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:--------------------------
PINION HARDNESS= 58.0HRC                WHEEL HARDNESS= 58.0HRC
PINION CONTACT ALLOWABLE=203000.0 psi   WHEEL CONTACT ALLOWABLE=203000.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi   WHEEL BENDING(UNIDIREC)= 56000.0 psi
MANUFACTURING METHOD=FINISH GROUND      AGMA QUALITY GRADE=11
========================================================================
```

GEAR DESIGN MAIN RESULT REPORT FOR 2ND REDUCTION

```
RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:----------------------------------------------------------
CENTRE DISTANCE=  401.000 MM            FACEWIDTH= 401.00 MM
NORMAL MODULE=14.0 MM                   HELIX ANGLE=12  9 43(12.1619, SINGLE)
NO. OF PINION TEETH= 28(  1 HUNTING)    NO. OF WHEEL TEETH=  28
PINION P.C.D.= 401.000 MM               WHEEL P.C.D.=  401.000 MM
PINION ADDN. MODIF. COEF.= 0.00000      WHEEL ADDN. MODIF. COEF.= 0.00000
---RATIOS:--------------------------------------------------------------
ACTUAL GEAR RATIO= 1.00(ERROR=  0.0%)   REQUIRED GEAR RATIO= 1.00
OVERLAP(FACE CONTACT) RATIO= 1.92       TRANSVERSE CONTACT RATIO=1.59
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):----------------------------
MESH POWER REQUIREMENT= 2280.0 KW       PINION RPM= 230.43
PINION PITTING RESISTANCE= 2332.9 KW    SERVICE FACTOR= 1.53( 1.50 REQUIRED)
WHEEL PITTING RESISTANCE= 2332.9 KW     SERVICE FACTOR= 1.53( 1.50 REQUIRED)
PINION BENDING STRENGTH= 2347.7 KW      SERVICE FACTOR= 1.54( 1.50 REQUIRED)
WHEEL BENDING STRENGTH= 2347.7 KW       SERVICE FACTOR= 1.54( 1.50 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:--------------------------
PINION HARDNESS= 58.0HRC                WHEEL HARDNESS= 58.0HRC
PINION CONTACT ALLOWABLE=203000.0 psi   WHEEL CONTACT ALLOWABLE=203000.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi   WHEEL BENDING(UNIDIREC)= 56000.0 psi
MANUFACTURING METHOD=FINISH GROUND      AGMA QUALITY GRADE=11
========================================================================
```

During the process of manual design of shaft components, it was found that some bearing OD's would interfere, after one or two days having been spent on preliminary calculations. However, this situation is quickly detected by the SHAFT DESIGN branch of the DRIVES package and warning messages are given to that effect. The interferences and conflicts in dimensions are also shown pictorially in the layout drawn by the parametric programs, as in Fig.8-7. For this particular

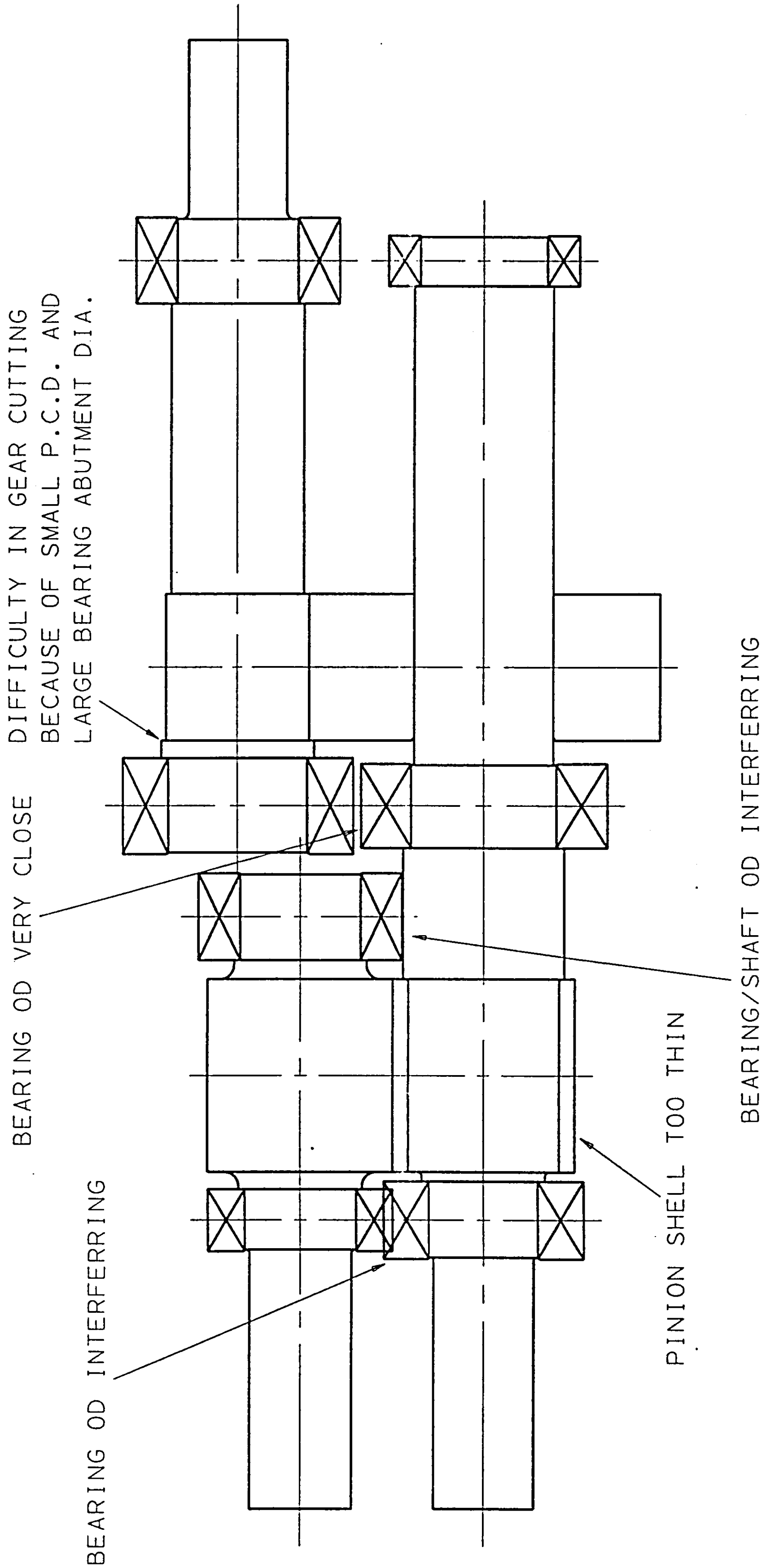


Fig.8-7 Layout of First Attempt Design for Cold Mill Box

design problem, the centre distances of gear sets are not determined by the gear capacity ratings, but rather by the bearing capacity ratings. The centre distances were eventually increased from 537 to 630 for the low speed gears and from 401 to 460 for the mill pinions. The final designs for the low speed gears(1ST REDUCTION) and mill pinions(2ND REDUCTION) are as follows.

```
            GEAR DESIGN MAIN RESULT REPORT FOR 1ST REDUCTION

RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:-----------------------------------------------------------
CENTRE DISTANCE=  630.000 MM          FACEWIDTH= 280.00 MM
NORMAL MODULE=18.0 MM                 HELIX ANGLE=12 50 53(12.8482, SINGLE)
NO. OF PINION TEETH= 19( 19 HUNTING)  NO. OF WHEEL TEETH=  48
PINION P.C.D.= 357.313 MM             WHEEL P.C.D.=  902.687 MM
PINION ADDN. MODIF. COEF.= 0.38000    WHEEL ADDN. MODIF. COEF.= 0.30000
---RATIOS:---------------------------------------------------------------
ACTUAL GEAR RATIO= 2.53(ERROR=  0.7%) REQUIRED GEAR RATIO= 2.51
OVERLAP(FACE CONTACT) RATIO= 1.10     TRANSVERSE CONTACT RATIO=1.42
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):-----------------------------
MESH POWER REQUIREMENT= 3800.0 KW     PINION RPM= 570.00
PINION PITTING RESISTANCE= 5012.1 KW  SERVICE FACTOR= 1.98( 1.50 REQUIRED)
WHEEL PITTING RESISTANCE= 5012.1 KW   SERVICE FACTOR= 1.98( 1.50 REQUIRED)
PINION BENDING STRENGTH= 4944.8 KW    SERVICE FACTOR= 1.95( 1.50 REQUIRED)
WHEEL BENDING STRENGTH= 5025.3 KW     SERVICE FACTOR= 1.98( 1.50 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:---------------------------
PINION HARDNESS= 58.0HRC              WHEEL HARDNESS= 58.0HRC
PINION CONTACT ALLOWABLE=203000.0 psi WHEEL CONTACT ALLOWABLE=203000.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi WHEEL BENDING(UNIDIREC)= 56000.0 psi
MANUFACTURING METHOD=FINISH GROUND    AGMA QUALITY GRADE=11
=========================================================================

            GEAR DESIGN MAIN RESULT REPORT FOR 2ND REDUCTION

RATED TO AGMA 6005-B89, ROLLING MILL SERVICE GEARS, 1989
---BASIC DATA:-----------------------------------------------------------
CENTRE DISTANCE=  460.000 MM          FACEWIDTH= 300.00 MM
NORMAL MODULE=16.0 MM                 HELIX ANGLE=10 55 27(10.9242, SINGLE)
NO. OF PINION TEETH= 28(  1 HUNTING)  NO. OF WHEEL TEETH=  28
PINION P.C.D.= 460.000 MM             WHEEL P.C.D.=  460.000 MM
PINION ADDN. MODIF. COEF.= 0.12000    WHEEL ADDN. MODIF. COEF.= 0.12000
---RATIOS:---------------------------------------------------------------
ACTUAL GEAR RATIO= 1.00(ERROR=  0.0%) REQUIRED GEAR RATIO= 1.00
OVERLAP(FACE CONTACT) RATIO= 1.13     TRANSVERSE CONTACT RATIO=1.54
---POWER RATINGS(INCLUSIVE SERVICE FACTORS):-----------------------------
MESH POWER REQUIREMENT= 2280.0 KW     PINION RPM= 225.63
PINION PITTING RESISTANCE= 2377.4 KW  SERVICE FACTOR= 1.56( 1.50 REQUIRED)
WHEEL PITTING RESISTANCE= 2377.4 KW   SERVICE FACTOR= 1.56( 1.50 REQUIRED)
PINION BENDING STRENGTH= 2362.2 KW    SERVICE FACTOR= 1.55( 1.50 REQUIRED)
WHEEL BENDING STRENGTH= 2362.2 KW     SERVICE FACTOR= 1.55( 1.50 REQUIRED)
---MATERIAL, MANUFACTURING AND PRECISION DATA:---------------------------
PINION HARDNESS= 58.0HRC              WHEEL HARDNESS= 58.0HRC
PINION CONTACT ALLOWABLE=203000.0 psi WHEEL CONTACT ALLOWABLE=203000.0 psi
PINION BENDING(UNIDIREC)= 56000.0 psi WHEEL BENDING(UNIDIREC)= 56000.0 psi
MANUFACTURING METHOD=FINISH GROUND    AGMA QUALITY GRADE=11
=========================================================================
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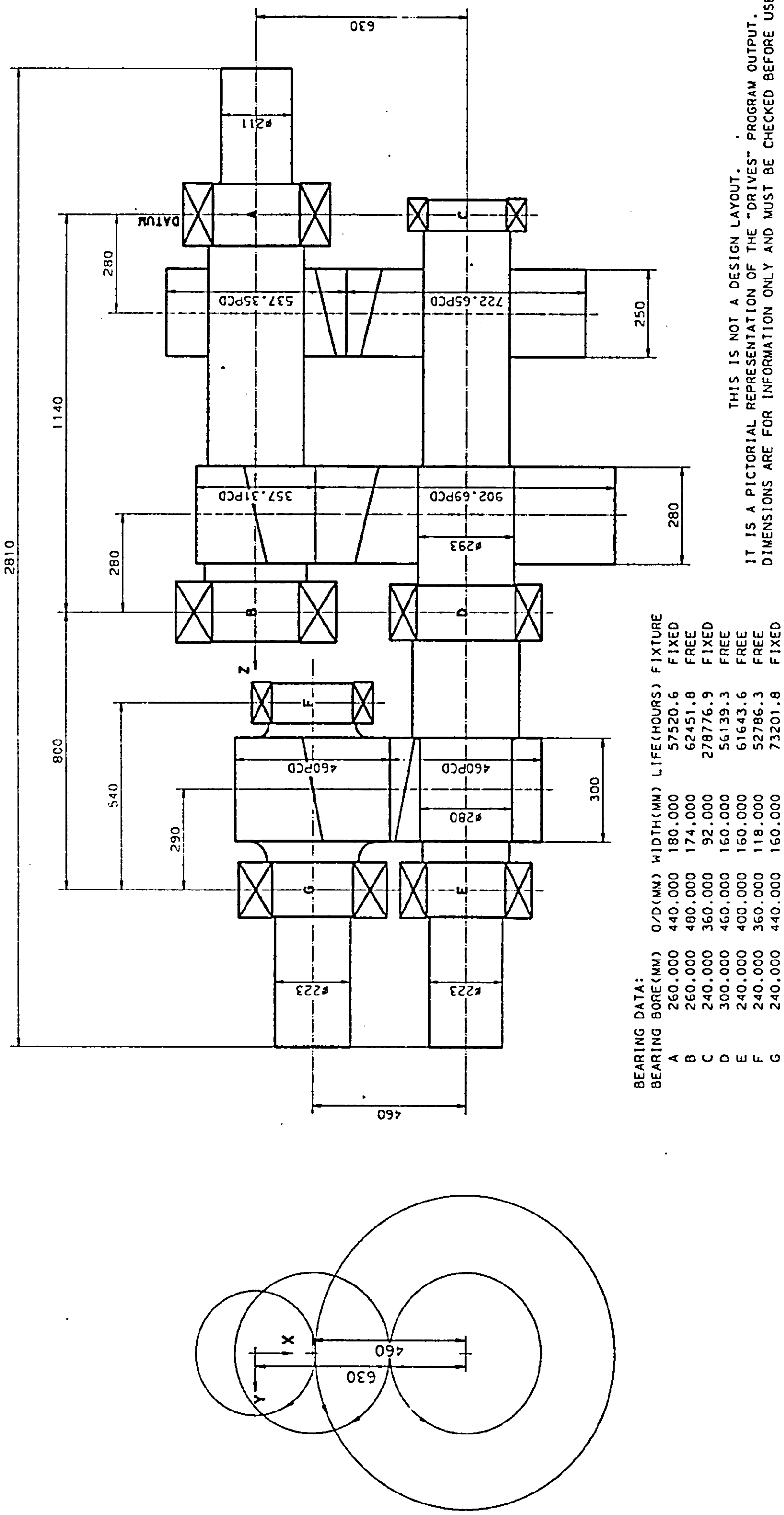


Fig.8-8 Layout of Final Design for Cold Mill Box

To reduce the surplus gear capacities introduced by the increased centre distance, the pinion aspect ratio and the face width were reduced. The face width of 280 for the low speed gears could not be further reduced because of the requirement of minimum face contact ratio being 1.1. The 'as built' design was arrived at after about a week's trial of repetitive use of the GEAR DESIGN part of the DRIVES to design gears at different centre distances, and long hand design of the shaft components including the bearings. In contrast to this, by using the GEAR DESIGN part and SHAFT DESIGN part interactively, together with the parametric programs, a very similar design, shown in Fig.8-8, is achieved in less than four hours time.

8.2.3 *Application Example 3*

Since the whole package of DRIVES has only been completed for a few months, the above use of the package for contract purpose has only exploited the GEAR DESIGN part, whilst the SHAFT DESIGN part and parametric programs are used afterwards for comparison. However, the whole package has been used for tender purposes recently, giving optimised designs and layouts in very short time periods. One good example is the design of the main drive gear boxes and mill pinions for a seven stand finishing mill, the overall arrangement being shown in Fig.8-9. Altogether, there were four reduction boxes, one speed up box, and seven mill pinion housings to be designed. Because of some earlier delays, the time left for designing the gear boxes was very limited. Fortunately, the DRIVES package had been completed and the designs of the

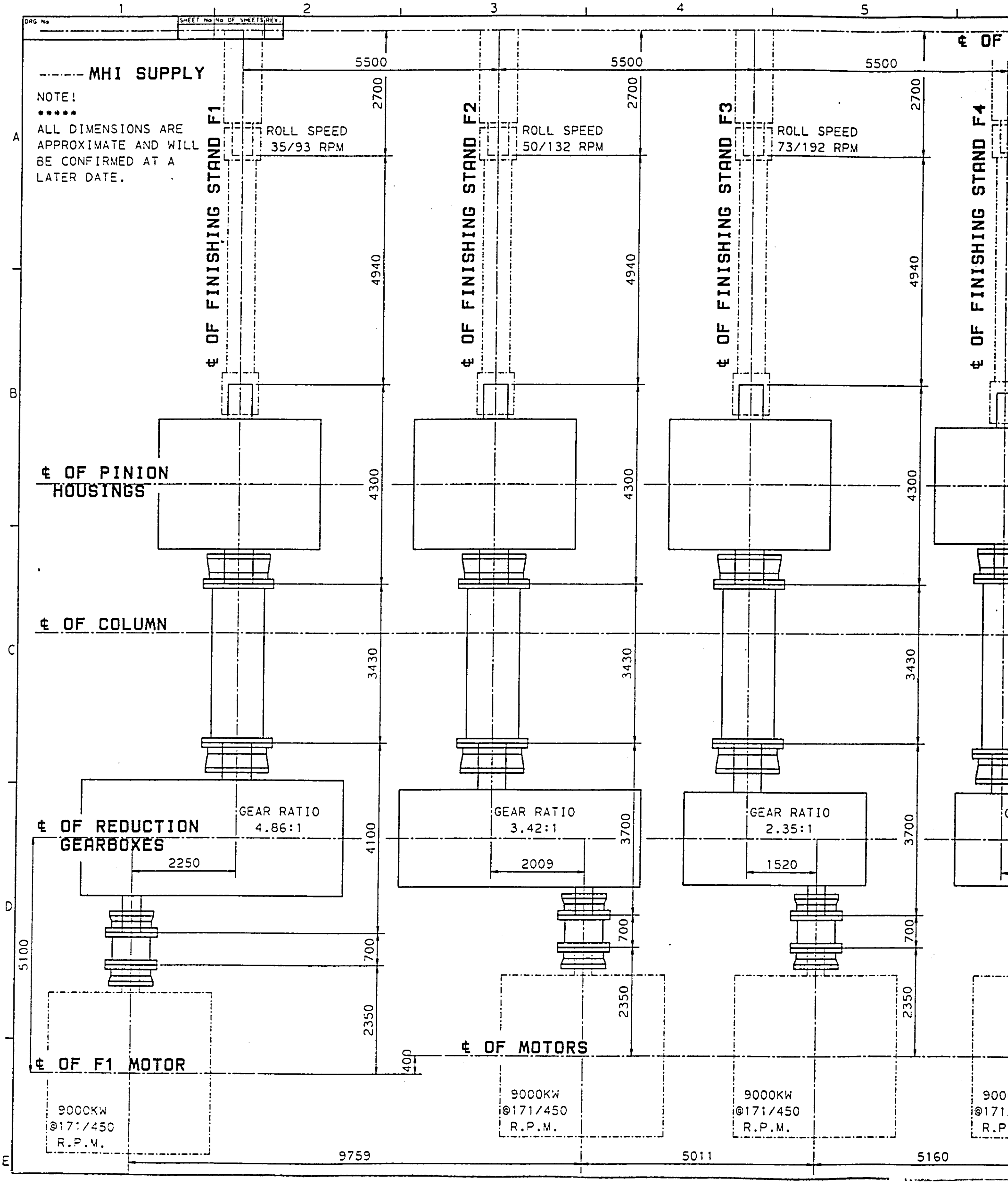


Fig.8-9 Schematic Layout of Hot Strip Mill Drives

MAIN DRIVE SYSTEM -REDUCTION GEARBOXES & COUPLINGS

01/04/94

	stand No	F1	F2	F3	F4	F5	F6	F7
main motor Kw & RPM		9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	7500 @ 218/588
main drive reducer								
gear ratio		4.86	3.42	2.35	1.33		1.16 S.U.	
Pinion								
type		double helical	single helical	single helical	single helical		single helical	
diameter		768.29	908.5	908.5	1011.43		1268.81	
No of teeth		21	26	26	33		50	
effective face		900	900	900	900		900	
material		17CrNiMo6	BS 970 826M40	BS 970 826M40	BS 970 826M40		BS 970 826M40	
heat treatment		CH&G	oil hardened and tempered	oil hardened and tempered	oil hardened and tempered		oil hardened and tempered	
hardness		58Rc	36Rc	36Rc	36Rc		36Rc	
Gear								
type		double helical	single helical	single helical	single helical		single helical	
diameter		3731	3110	2131.49	1348.57		1091.18	
No of teeth		102	89	69	44		43	
effective face		900	900	900	900		900	
material		BS 970 826M40	BS 970 826M40	BS 970 826M40	BS 970 826M40		BS 970 826M40	
heat treatment		oil hardened and tempered	oil hardened and tempered	oil hardened and tempered	oil hardened and tempered		oil hardened and tempered	
hardness		34Rc	34Rc	34Rc	34Rc		34Rc	
backlash		to be determined	to be determined	to be determined	to be determined		to be determined	
accuracy		AGMA Gr 11	AGMA Gr 11	AGMA Gr 11	AGMA Gr 11		AGMA Gr 11	
Bearings								
maker		SKF or equivalent	SKF or equivalent	SKF or equivalent	SKF or equivalent		SKF or equivalent	
type		double row spherical	double row spherical	double row spherical	double row spherical		double row spherical	
input motor		420/700/224	420/700/224	420/700/224	420/700/224		420/700/224	
life in hours		82834	82834	82834	119206		103341	
input mill		420/700/224	500/830/325	500/830/325	460/760/300		340/580/243	
life in hours		82834	101614	101616	106440		79517	
output motor		380/620/194	380/560/180	380/620/194	420/620/200		400/600/200	
life in hours		124702	95973	116740	89750		106286	
output mill		710/1030/236	630/920/290	560/920/280	460/760/300		400/720/256	
life in hours		1132654	86800	89814	122582		99516	
service factors @ 375rpm								
pinion strength		6.0225	5.2998	5.2341	5.1465		5.0589	
wheel strength		5.0589	5.4531	5.2341	5.037		5.3436	
pinion wear		7.9059	4.818	4.2924	4.2486		4.161	
wheel wear		3.9201	4.38	3.8763	3.8544		4.9932	
service factors @ 171rpm								
pinion strength		2.75	2.42	2.39	2.35		2.31	
wheel strength		2.31	2.49	2.39	2.3		2.44	
pinion wear		3.61	2.2	1.96	1.94		1.9	
wheel wear		1.79	2	1.77	1.76		2.28	
GD^2 of complete mill system referred to motor shaft		insufficient information to calculate	insufficient information to calculate	insufficient information to calculate	insufficient information to calculate	insufficient information to calculate	insufficient information to calculate	insufficient information to calculate

Table 8-1 Reduction Gear Boxes for Hot Strip Mill

MAIN DRIVE SYSTEM -PINION HOUSINGS

01/04/94

	stand No	F1	F2	F3	F4	F5	F6	F7
main motor Kw & RPM		9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	9000 @ 171/450	7500 @ 218/588
main drive pinion housing								
gear ratio		1:1	1:1	1:1	1:1	1:1	1:1	1:1
pinion								
type		double helical	double helical	double helical	double helical	double helical	double helical	double helical
construction		shell	shell	shell	integral	integral	integral	integral
diameter		1000	1000	1000	790	790	790	790
No of teeth		30	30	30	25	25	25	25
effective face		1600	1600	1600	1300	1300	1300	1300
shell material		17CrNiMo6	17CrNiMo6	17CrNiMo6	-	-	-	-
shaft material		BS 970 826M40	BS 970 826M40	BS 970 826M40	BS 970 826M40	BS 970 826M40	BS 970 826M40	BS 970 826M40
hardness		58Rc	58Rc	58Rc	36Rc	36Rc	36Rc	36Rc
backlash		to be determined	to be determined	to be determined	to be determined	to be determined	to be determined	to be determined
accuracy		AGMA Gr 11	AGMA Gr 11	AGMA Gr 11	AGMA Gr 11	AGMA Gr 11	AGMA Gr 11	AGMA Gr 11
heat treatment		CH&G	CH&G	CH&G	oil hardened and tempered	oil hardened and tempered	oil hardened and tempered	oil hardened and tempered
Bearings								
maker		SKF or equivalent	SKF or equivalent	SKF or equivalent	SKF or equivalent	SKF or equivalent	SKF or equivalent	SKF or equivalent
type		double row spherical	double row spherical	double row spherical	double row spherical	double row spherical	double row spherical	double row spherical
input motor		630/920/290	630/920/290	630/920/290	420/620/200	420/620/200	420/620/200	420/620/200
life in hours		134386	>F1	>F1	133039	>F4	>F4	>F4
input mill		560/920/280	560/920/280	560/920/280	360/600/192	360/600/192	360/600/192	360/600/192
life in hours		163755	>F1	>F1	133039	>F4	>F4	>F4
output motor		460/760/300	460/760/300	460/760/300	320/540/218	320/540/218	320/540/218	320/540/218
life in hours		118440	>F1	>F1	133039	>F4	>F4	>F4
output mill		560/920/280 -	560/920/280	560/920/280	360/600/192	360/600/192	360/600/192	360/600/192
life in hours		163755	>F1	>F1	133039	>F4	>F4	>F4
service factors @ 375rpm assuming 60% split of torque								
pinion strength		5.1465	7.3365	10.4244	5.3874	7.2489	8.4315	10.9938
pinion wear		4.8837	6.9423	9.855	3.9639	5.3217	6.1977	8.0811
service factors @ rpm assuming 50% split of torque		34.9	50	71.84	125.7	171	200	218
pinion strength		2.82	4.02	5.712	2.952	3.972	4.62	6.024
pinion wear		2.676	3.804	5.4	2.172	2.916	3.396	4.428
service factors @ rpm assuming 60% split of torque		34.9	50	71.84	125.7	171	200	218
pinion strength		2.35	3.35	4.76	2.46	3.31	3.85	5.02
pinion wear		2.23	3.17	4.5	1.81	2.43	2.83	3.69
motor couplings								
size		GO-10BSHT	GO-10BSHT	GO-10BSHT	GO-10BSHT	AO-10BSHT	GO-10BSHT	AO-10BSHT
type		gear type - shear pin	gear type - shear pin	gear type - shear pin	gear type - shear pin	gear type - shear pin	gear type - shear pin	gear type - shear pin
maker		Maina or similar	Maina or similar	Maina or similar	Maina or similar	Flender or similar	Maina or similar	Flender or similar
torque rating in N-m		1670000	1670000	1670000	1670000		1670000	
lead spindles								
maker		Maina or similar	Maina or similar	Maina or similar	Maina or similar		Maina or similar	
type		gear type	gear type	gear type	gear type		gear type	
size		AO-16BHT	AO-16BHT	AO-16BHT	AO-12BHT		AO-12BHT	
length - approx.		3610	3610	3610	4500	6700	4500	6700
flange diameter		1540	1540	1540	1292	1112	1292	1112
torque rating in N-m		8150000	8150000	8150000	2330000	1670000	2330000	1670000

Table 8-2 Mill Pinion Housings for Hot Sript Mill

boxes and mill pinion housings were done in four days, which would have taken four to six weeks without the package, according to the estimates of engineers at the collaborating establishment.

The designs are summarised in Table 8-1 for the reduction boxes and Table 8-2 for the mill pinion housings. Parametric layout drawings for the reduction gear box and mill pinion housing of stand F1 are shown in Fig.8-10 and 8-11. The service factors required are K_{SF}=2.25 for the bending strength and C_{SF}=1.75 for the pitting resistance(wear). Comparing the service factors of Table 8-1 for the reduction boxes, it can be found that the limiting factor for the gears is the gear wheel pitting resistance(wear), with the exceptions of F2 and F6 gears. These two exceptions in service factors are caused by consideration of having some parts interchangeable to reduce the cost of engineering, manufacturing and maintenance. For the four reduction boxes for stands F1 to F4, all the pinion loads are the same(9000 kw @ 171 rpm) and the gear materials could be chosen the same. The difference is in the gear ratios, which result in different values for the geometry factor I and hence different pitch circle diameters of pinion. However, the pinion sizes of stands F2 and F3 are considered similar enough to be made interchangeable, which makes F2 gears slightly over designed for gear capacities. The centre distances of the gear boxes for stands F4 and F6 are also found to be similar and made interchangeable, which results in F6 gears over designed for gear capacities. For the mill pinion housings, stands F1 to

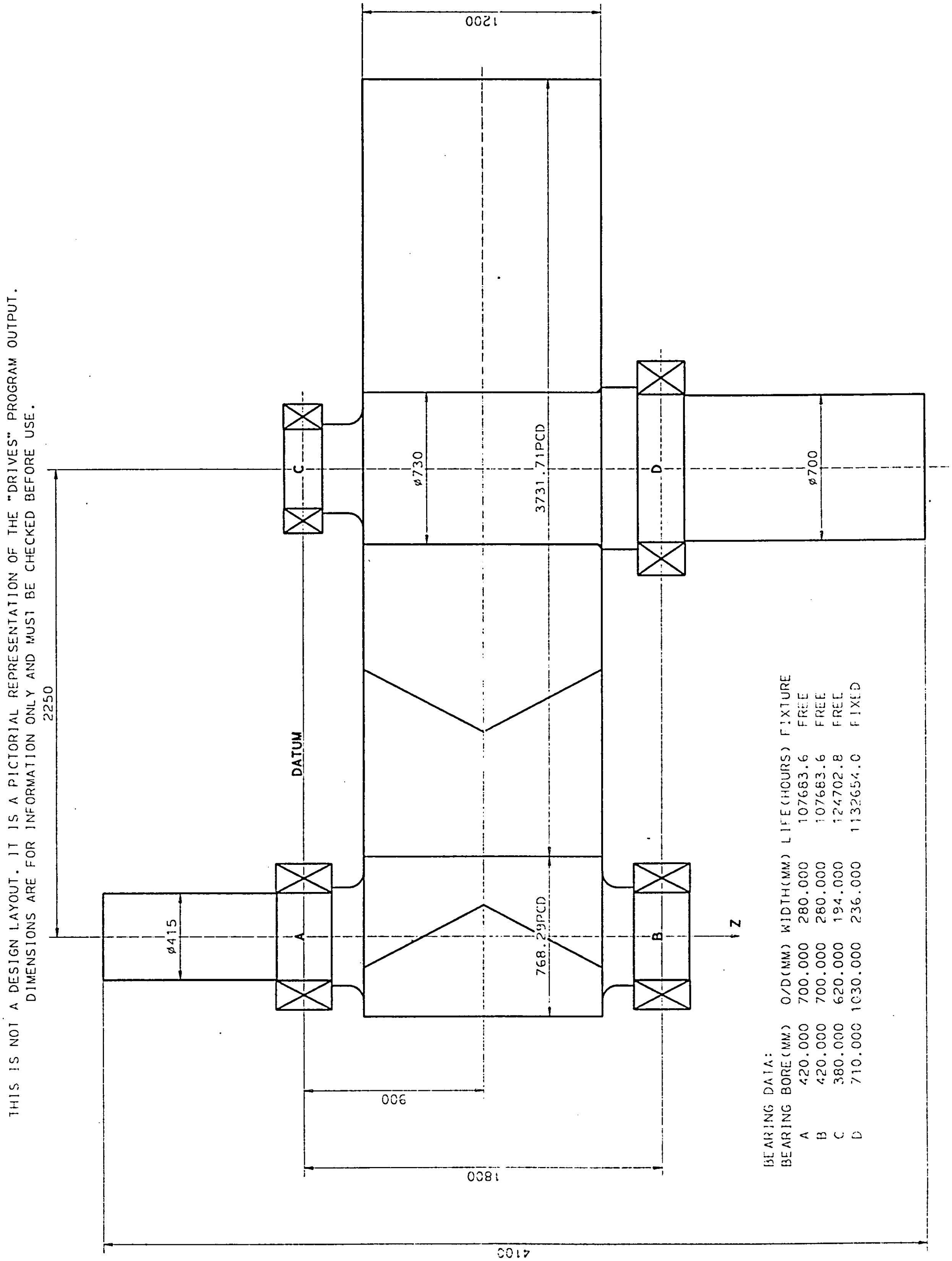


Fig.8-10 Layout of Stand F1 Reduction Box

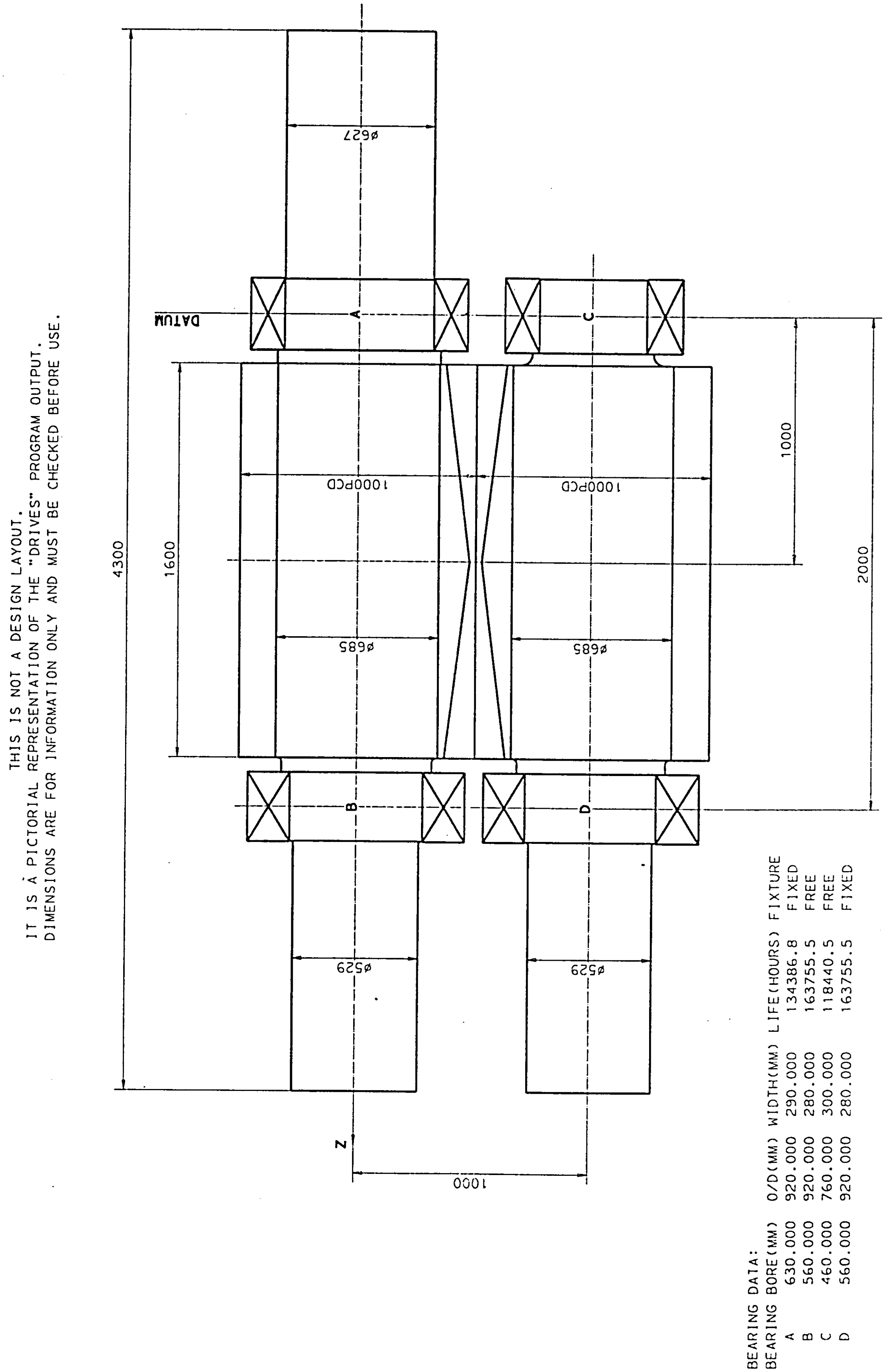


Fig.8-11 Layout of Stand F1 Mill Pinion Housing

F3 are made the same on the basis of the design for F1 stand, and stands F4 to F7 are made the same on the basis of the design for F4 stand. The service factors for F1 mill pinions show that they are over designed for gear capacities. The reason for the over design is as follows. The input mill pinion shaft shown in Fig.8-11 is 4300 mm long, which makes it impossible for the gear teeth to be heat treated if integral pinion shaft is used. For the shell pinion structure as shown in Fig.8-11, the pinion root diameter has to be larger than the minimum shaft diameter determined by stresses plus twice the minimum shell thickness. The pitch circle diameter so determined is larger than the minimum pitch circle diameter determined by gear capacities, which results in the over design of gear capacities.

9.1 *Discussion*

The DRIVES package has proven to be a very useful design tool for the collaborating establishment, as shown by the application examples in Chapter Eight. In the following, the achievements and limitations of this investigation are discussed and suggestions on further work are made where appropriate, but summarised in Section 9.3.

For a numerical optimisation problem, such as optimum gear design, there are two parts of it, viz., the optimum design model and optimum design algorithm. The main aim of this investigation is the development of an algorithm for the optimum design of gears. The algorithm developed has been shown, as by the example problem of Chapter Four and the analytical example of Chapter Eight, to be efficient and effective in finding the optimum gear design. The efficiency of the algorithm is shown by the small number of capacity evaluations required to find the optimum design, with typical computations ranging from 30 to 150 capacity evaluations. Since all geometrically feasible designs are exhausted at a centre distance, and the centre distance is minimised using the macro monotonic and micro non-monotonic relations, the algorithm is also effective in that the optimum design is the global optimum. In example 1 of Industrial Application in Chapter Eight, indicative figures show that the savings in weight could be around 10% to 20%, compared with conventional design methods.

The optimum design obtained by the algorithm is the optimum under the constraints considered. However, when the DRIVES package is used to solve real industrial application problems, many other considerations may come into play and the design used may not be the optimum gear design. In example 2 of Industrial Application in Chapter Eight, it is shown that the centre distance is not determined by the gear capacities, but rather by the bearing capacities. This demonstrates the importance of integrating the design of gears and design of shaft components in one software package. The level of integration that DRIVES has achieved is data integration by the mechanism of Fortran COMMON data bank and it helps to <u>detect</u> any conflicts in the design of gears and the design of shaft components. However, integration at model level would <u>avoid</u> any conflicts in designs happening and help to achieve a system optimum. The integration at model level requires the optimum design model to be expanded to include constraints for shaft components and makes the whole optimisation problem much more complicated, especially for multiple stage gear boxes. It would be desirable if further work could be done in this respect. The work of Savage et al. [41] is a good starting reference, although it only dealt with single stage box of spur gears. From the experience of using the DRIVES package at the collaborating establishment, it seems that gear design and shaft component design are rarely in conflict for reduction gear boxes. Occasionally, it may happen that input pinion root diameter is larger than the bearing abutment diameter, as shown in Fig.8-7, causing difficulty in gear cutting. For mill pinions and compact drives, bearing OD interference at the minimised gear centre distance is more common, the reason being that the gear ratio is 1 to 1 and wider pinion aspect ratio is used to make the gear centre distance similar to the centre distance of the working rolls of the mill.

The DRIVES package is capable of dealing with the design of multiple stage gear boxes but the optimisation algorithm is for single gear set. The division of overall gear ratio for the stages has not been dealt with directly in this thesis. Willis [52] and ESDU 88033 [31] have proposed formulae for the division of the overall gear ratio for gears with the same material, on the basis of minimising total rotor wight or volume, as shown in equations (2-4-1) to (2-4-5). At the collaborating establishment, the formulae used for the division of the overall gear ratio for a double reduction box

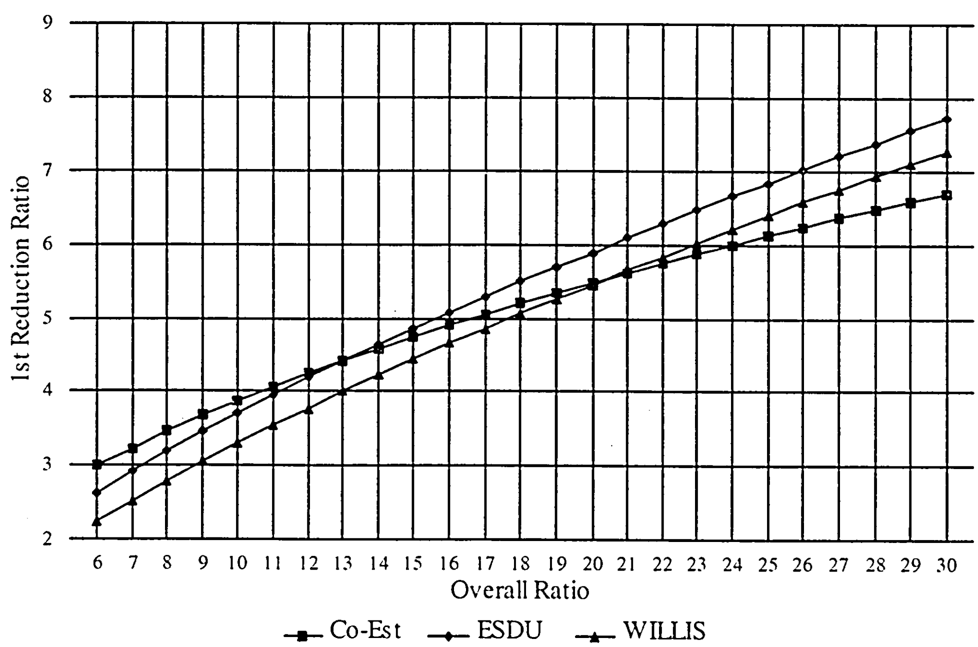
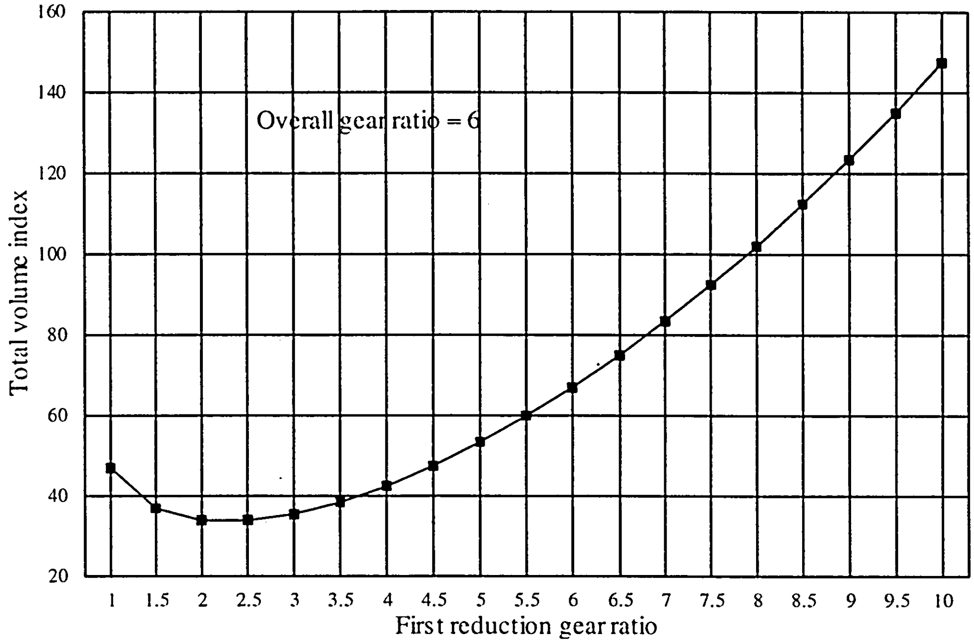
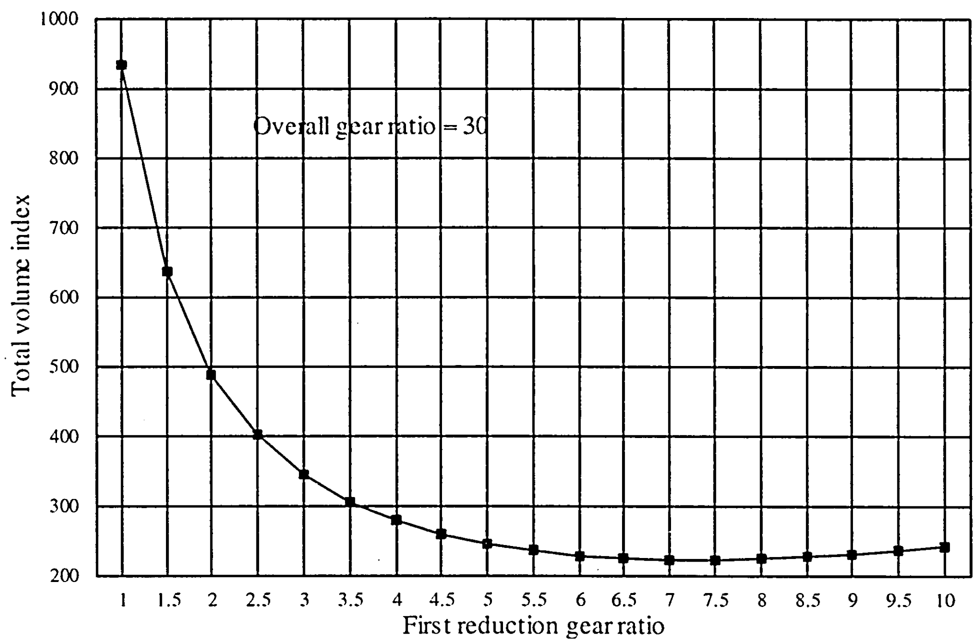


Fig.9-1 Division of gear ratio for double reduction box

are given in equations (8-2-1) and (8-2-2). Fig.9-1 shows a comparison of the first reduction gear ratios given by the formulae of Willis, ESDU and the collaborating establishment. It can be seen that the Co-Est curve favours higher first reduction ratio for small overall gear ratio and lower first reduction ratio for large overall gear ratio, but the difference is not very large. Fig.9-2a and b show the sensitivity of total rotor weight to the first reduction gear ratio, using the volume index of the double reduction gears given by equation (2-4-1) by Willis. It can be seen that the total volume index is not sensitive to the variation in the first reduction gear ratio around the optimum point. Considering the difference in first reduction gear ratios shown in Fig.9-1 and the sensitivity of the volume index to the first reduction gear ratio as shown in Fig.9-2, the formulae given by Willis, ESDU and the collaborating establishment will not result in significant difference in total rotor weight. Thus the formulae of the collaborating establishment are used in the DRIVES for their simplicity and conventional usage. However, further work could be carried out for the optimum division of overall gear ratio when gear materials are not the same in the gear train, which is quite common in real industrial application. Two different approaches may be taken for the further investigation. One is the analytical approach similar to that taken by Willis to derive formulae for the division of gear ratios with different gear materials, which can then be used by an algorithm dealing with the design of single gear set.



(a)



(b)

Fig.9-2 Relative weight expressed by volume index

The alternative is a numerical approach which treats the gear ratio of each stage as a variable in the optimum design model and changes the objective to minimising total rotor volume or overall centre distance. It should be born in mind however, that the division of overall gear ratio can affect the situation concerning conflicts in the designs of shaft components and gears. Thus if the alternative numerical approach for gear ratio division can be combined with the integrated model for gear design and shaft component design, a comprehensive system optimisation model is obtained. The problem is then to devise the algorithm for solving this complex model, which will yield a true system optimum for the design of the gear box but will be very difficult indeed.

Through the study of the AGMA standards for gear capacity rating, especially the study on geometry factors I and J, a better understanding has been achieved of the numerical behaviour of the AGMA gear capacity rating in relation to geometric changes, such as changes in centre distance, face width, helix angle, etc. One example is the micro non-monotonicity of power capacity rating, as shown in Fig.4-13, where a small increase in centre distance may cause a reduction in power capacity rating, which can be explained by the unfavourable change in the load sharing ratio. The reciprocal and trigonometric relations of the form factor and stress correction factor to the number of teeth and helix angle are also useful in that close form formulae can be developed for the calculation of geometry factor J for standardised tooth systems in place of the iterative numerical procedure provided by AGMA. However, there are many other factors that can affect the capacity of a gear set and only the major ones have been considered in gear standards. Even for these major factors, there are uncertainties in their values, which means that the gear capacities given by gear standards are only predictions and should be taken with caution. Having said that, gear standards are the one best tool available to a designer to assess the capacity of a gear set at the design stage, if a prototype is too expensive. Often, the numerical results are the only basis for discussion between the designer and user of a gear box.

The optimisation algorithm developed in this thesis is based on AGMA standards and similar work can be done on the basis of other gear rating standards, such as BS and ISO standards. The gear rating methods specified in BS and ISO standards are different from the AGMA methods and the algorithm cannot be applied directly to the BS and ISO standards. A new investigation on the local or micro behaviour of the power capacity rating would have to be carried out and new boundaries for the search interval for the centre distance would have to be established. However, the strategy for one dimensional search based on the general or macro monotonicity would still be valid. The method for finding the maximum power combination at a fixed centre distance could also be the same, since it is based on geometric constraints which would still apply for gears designed to BS or ISO standards.

On the practical side, the parametric programs have proven to be a most useful part of the software package in helping the design engineers to evaluate the designs produced by the DRIVES. Any conflict in dimensions and the relative sizes of the shaft and bearings in relation to the gears are all shown in the layout drawn by the parametric programs. Because of the interactive way that the DRIVES and parametric programs operate, it is very easy for the design engineers to switch to one window to change the design and then to another window to see the effect of the change in the layout drawing. The descriptive layout definition by centre vectors has also been a success. Because of the flexibility with which the user can set up the coordinate system and define the centre vectors, complex layouts such as that shown by Fig.5-1a can be defined to the DRIVES and parametric programs as easily as a simple layout such as that of Fig.5-1b. The menu driven user interface also contributes to the easiness with which the DRIVES and parametric programs can be used, which in conjunction with the efficient algorithms for gear design and shaft component design, helps to reduce the design time very significantly, as demonstrated by example 3 of Industrial Application in Chapter Eight.

9.2 *Conclusions*

This investigation has been successful in developing efficient and effective optimisation algorithms for gear design, and integrating gear design with shaft design into one software package together with parametric layout drawing generation.

The objective of the optimum gear design is to minimise the gear centre distance which is the main index of the size of a gear box. The algorithm for minimising the centre distance is based on transforming the design constraints into direct limiting boundaries on design variables, and the macro monotonic and micro non-monotonic behaviour of power capacity rating. In other words, gear strength constraints are transformed into lower and upper boundaries on the search interval for the centre distance, using maximum and minimum limiting values of geometry factors. The centre distance is minimised between the boundaries, using the monotonic property to get increasingly closer boundaries according to the maximum power capacity of a trial value of the centre distance. The transformation of geometry constraints such as minimum face contact ratio, minimum number of pinion teeth, maximum helix angle, and gear ratio tolerance, results in a fairly small feasible domain for the combination of module, numbers of teeth of pinion and gear. The local search finally eliminates the effect of the micro non-monotonicity and finds the global minimum design for the centre distance. An example in industrial application shows significant savings in weight or volume of the gear set when compared with conventional design methods.

The descriptive definition of layout by centre vectors is a versatile frame for reaction calculation and dimension conflict detection. Shaft and bearing design is based on established theories for reaction calculation, bearing life rating, shaft stress calculation and shaft failure criterion. For the bearing reaction calculation, a finite element model for beams based on simple plane bending theory is used, which is capable of dealing with three bearing shafts with different section diameters.

A well defined data organisation and data file provide the link between the calculating programs and the parametric programs. The descriptive layout definition and data file are used to draw the individual shafts, bearings, and gears in the parametric layout and to put the main dimensions on the drawing.

The contribution to knowledge by this investigation is mainly in the gear design area. Through the study of the macro and micro properties of the power capacity rating, the numerical behaviour of AGMA standards in relation to gear geometry is clearly revealed. This knowledge of the numerical behaviour can help the design engineer to have a better understanding of the design results and the effect on the design of changes in gear geometry. The optimisation algorithm establishes a logical procedure for the design of minimum centre distance gears and is itself partly based on this knowledge of the numerical behaviour of the AGMA standards. Other novel features of the algorithm are: the strategy of breaking down the optimisation problem into minimisation of centre distance and maximisation of power capacity at a given centre distance, and transformation of the gear design constraints into direct boundaries on design variables. The descriptive layout definition by centre vectors is also a novel and flexible way to specify the geometric relations of components in gear boxes.

The final product of this investigation is the software package DRIVES and the parametric programs. Although the optimum design initially given by the DRIVES is not always used in real industrial applications, it is nevertheless a good base to start a design in the sense that any other evolved design must be larger than the optimum design. The package DRIVES, together with the parametric programs have proven to be a very useful design tool at the collaborating establishment and to the author's knowledge, no other design software package to date has achieved this level of system integration of optimum gear design with shaft design and parametric layout.

9.3 *Recommendations for Further Work*

9.3.1. *Optimisation of the overall system by including shaft and bearing design constraints in the optimum design model*

This extension of the optimum design model will introduce considerable complexity to the problem, especially for multiple stage gear boxes. The first question to be asked will be what the objective function should be for such a system problem. Rotor volumes of gears, shafts and bearings in combination with the weighted volume of a tight box wraping all the components may serve as an objective function for a first attempt. A special purpose algorithm for this problem will be very difficult to devise indeed. Savage et al. [41] used a general purpose algorithm for solving a single stage problem and can serve the purpose of a good starting reference.

9.3.2 *Optimum division of the overall gear ratio for a multiple stage gear box*

The work can be done by considering gears of different materials in the gear train. An analytical approach similar to that of Willis [52] can be taken, but employed for different materials. Alternatively, a numerical approach may be taken in which the gear ratios of individual stages are treated as design variables. This may mean another level of optimisation of the overall volume or centre distance in addition to the two levels of optimisation proposed in this thesis.

9.3.3 *Investigation of volume minimisation by including pinion aspect ratio as a design variable*

The investigation may be conducted by carrying out some numerical experiments to see if any functional relations can be established between the minimised volume and pinion aspect ratio. In the numerical experiments, the minimised volume for a given aspect ratio may be found by using the optimum design method developed in this thesis.

9.3.4 *Optimum gear design based on non-AGMA Standards*

For application to another gear rating standard, such as BS or ISO, a new investigation on the local or micro behaviour of the standard would have to be carried out and a new search interval for the centre distance be established. The strategy for the one dimensional search for the minimum centre distance and the method for finding the maximum power combination could remain the same.

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Appendix A Derivation of Transverse Contact Ratio

Transverse contact ratio m_p is the ratio of the active length of line of action to the transverse base pitch p_b. Fig.A-1 is a diagram showing the relevant points along the line of action. These points are:

- Point a, the tangential point of line of action with the base circle of pinion;
- Point b, the intersecting point of line of action with the tip circle of gear;
- Point c, the intersecting point of line of action with the tip circle of pinion;
- Point d, the tangential point of line of action with the base circle of gear.

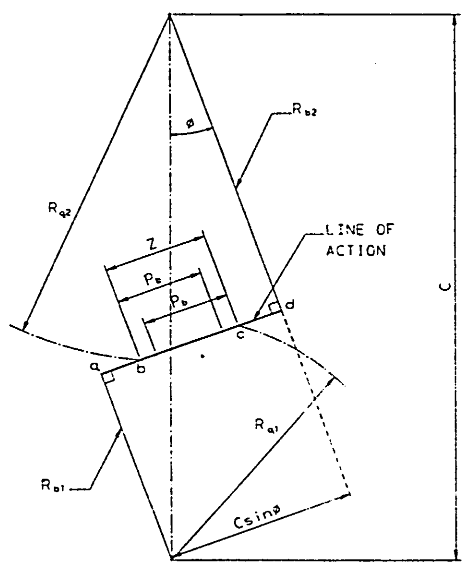

Fig.A-1 Transverse view of the line of action

In the following derivation, the length of a line segment is designated by the start and end points of the line segment. For example, the length of the line segment between points a and b is ab. Thus the active length of line of action Z is bc and transverse contact ratio is

$$m_p = \frac{bc}{p_b} \tag{A-1}$$

From Fig.A-1, the following equations can be established:

$$ac = ab + bc \tag{A-2}$$

$$bd = bc + cd \tag{A-3}$$

$$ad = ab + bc + cd \tag{A-4}$$

Comparing (A-4) with (A-2) and (A-3), it can be seen that

$$ac + bd = ad + bc \tag{A-5}$$

or

$$bc = ac + bd - ad \tag{A-6}$$

The lengths of line segments ac, bd and ad can be calculated by the following equations, based on Fig.A-1,

$$ac = (R^2_{a1} - R^2_{b1})^{1/2} \tag{A-7}$$

$$bd = (R^2_{a2} - R^2_{b2})^{1/2} \tag{A-8}$$

$$ad = C \sin\phi \tag{A-9}$$

where

$$R_{a1} = \frac{m_n N_1}{2\cos\psi} + h_a m_n \tag{A-10}$$

is the tip circle radius of pinion,

$$R_{b1} = \frac{m_n N_1}{2\cos\psi}\cos\phi \tag{A-11}$$

is the base circle radius of pinion,

$$R_{a2} = \frac{m_n N_2}{2\cos\psi} + h_a m_n \tag{A-12}$$

is the tip circle radius of gear,

$$R_{b2} = \frac{m_n N_2}{2\cos\psi}\cos\phi \qquad \text{(A-13)}$$

is the base circle radius of gear, and

$$C = \frac{m_n}{\cos\psi}\,\frac{N_1 + N_2}{2} \qquad \text{(A-14)}$$

is the centre distance. Equations (A-10) through (A-14) are valid for gear sets without addendum modifications.

Substituting equations (A-10) through (A-14) into equations (A-7), (A-8) and (A-9) gives

$$ac = \frac{m_n}{\cos\psi}[(\frac{N_1}{2} + h_a\cos\psi)^2 - (\frac{N_1}{2}\cos\phi)^2]^{1/2} \qquad \text{(A-15)}$$

$$bd = \frac{m_n}{\cos\psi}[(\frac{N_2}{2} + h_a\cos\psi)^2 - (\frac{N_2}{2}\cos\phi)^2]^{1/2} \qquad \text{(A-16)}$$

$$ad = \frac{m_n}{\cos\psi}\,\frac{N_1 + N_2}{2}\cos\phi \qquad \text{(A-17)}$$

Again, substituting equations (A-15), (A-16) and (A-17) into equation (A-6) and noticing $m_G = N_2/N_1$, gives

$$bc = \frac{m_n N_1}{\cos\psi}\{[(\frac{1}{2} + \frac{h_a\cos\psi}{N_1})^2 - (\frac{1}{2}\cos\phi)^2]^{1/2} + [(\frac{m_G}{2} + \frac{h_a\cos\psi}{N_1})^2 - (\frac{m_G}{2}\cos\phi)^2]^{1/2} - \frac{1+m_G}{2}\sin\phi\} \qquad \text{(A-18)}$$

The transverse base pitch p_b is

$$p_b = \frac{m_n}{\cos\psi}\pi\cos\phi \qquad \text{(A-19)}$$

Substituting equations (A-18) and (A-19) into equation (A-1) gives the transverse contact ratio

$$m_p = \frac{N_1}{\pi\cos\phi}[[(\frac{1}{2} + \frac{h_a\cos\psi}{N_1})^2 - (\frac{\cos\phi}{2})^2]^{1/2} +$$

$$+ [(\frac{m_G}{2} + \frac{h_a\cos\psi}{N_1})^2 - (\frac{m_G\cos\phi}{2})^2]^{1/2} - \frac{1+m_G}{2}\sin\phi\} \qquad \text{(A-20)}$$

Appendix B Geometrically feasible combinations

for the Numerical Example in Chapter Four

$P_{min} = \min\{P_{ac1}, P_{ac2}, P_{at1}, P_{at2}\}$

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
700	1022	269	12	22	91	14.403	0.2120	-1.5	1.77	1.59	2073	1027	1389	1022
700	991	269	14	19	79	11.478	0.2471	-1.0	1.22	1.58	2000	991	1496	1082
700	952	269	12	22	92	12.274	0.2222	-0.4	1.52	1.61	1953	967	1300	952
700	938	269	12	22	93	9.696	0.2274	0.6	1.20	1.63	1952	967	1280	938
700	886	269	10	26	110	13.729	0.1924	0.7	2.03	1.63	2064	1022	1195	886
700	851	269	10	26	111	11.883	0.1926	1.6	1.76	1.65	1992	987	1145	851
700	807	269	9	29	123	12.274	0.1774	1.0	2.02	1.67	2061	1021	1082	807
700	783	269	9	29	122	13.901	0.1723	0.2	2.29	1.65	1989	985	1049	783
700	770	269	9	29	124	10.400	0.1826	1.8	1.72	1.68	1985	983	1033	770
700	737	269	9	30	124	8.110	0.1769	-1.6	1.34	1.70	1966	974	987	737
701	1035	270	12	22	91	14.718	0.2120	-1.5	1.82	1.59	2100	1040	1407	1035
701	985	270	14	19	79	11.874	0.2471	-1.0	1.26	1.58	1988	985	1490	1077
701	971	270	12	22	92	12.644	0.2172	-0.4	1.57	1.61	1982	982	1320	971
701	930	270	12	22	93	10.164	0.2274	0.6	1.26	1.63	1931	956	1270	930
701	884	270	10	26	110	14.060	0.1874	0.7	2.09	1.63	2052	1017	1188	884
701	864	270	10	26	111	12.265	0.1926	1.6	1.83	1.65	2020	1000	1163	864
701	802	270	9	29	123	12.644	0.1774	1.0	2.09	1.66	2046	1013	1076	802
701	783	270	9	29	124	10.836	0.1826	1.8	1.80	1.68	2015	998	1051	783
701	780	270	9	29	122	14.227	0.1723	0.2	2.35	1.65	1982	981	1045	780
701	757	270	9	30	124	8.664	0.1770	-1.6	1.44	1.70	2013	997	1015	757
702	982	270	12	22	92	13.003	0.2172	-0.4	1.61	1.60	2005	993	1337	982
702	978	270	14	19	79	12.256	0.2471	-1.0	1.30	1.58	1975	978	1483	1072
702	923	270	12	22	93	10.609	0.2224	0.6	1.32	1.62	1908	945	1257	923
702	879	270	10	26	110	14.382	0.1874	0.7	2.13	1.63	2042	1011	1182	879
702	873	270	10	26	111	12.635	0.1926	1.6	1.88	1.64	2039	1010	1175	873
702	818	270	10	27	112	8.098	0.1920	-1.2	1.21	1.68	2004	992	1101	818
702	796	270	9	29	123	13.003	0.1774	1.0	2.15	1.66	2031	1006	1068	796
702	794	270	9	29	124	11.255	0.1776	1.8	1.86	1.68	2035	1008	1063	794
702	786	270	9	29	122	14.545	0.1723	0.2	2.40	1.65	1996	989	1053	786
702	774	270	9	30	124	9.184	0.1720	-1.6	1.52	1.70	2045	1013	1035	774
703	993	270	12	22	92	13.351	0.2172	-0.4	1.65	1.60	2027	1004	1352	993
703	973	270	14	19	79	12.626	0.2471	-1.0	1.34	1.57	1964	973	1476	1067
703	917	270	12	22	93	11.036	0.2224	0.6	1.37	1.62	1890	936	1249	917
703	882	270	10	26	111	12.994	0.1926	1.6	1.93	1.64	2058	1019	1188	882
703	875	270	10	26	110	14.697	0.1874	0.7	2.18	1.62	2032	1006	1177	875
703	805	270	10	27	112	8.652	0.1920	-1.2	1.29	1.68	1964	973	1083	805
703	803	270	9	29	124	11.657	0.1776	1.8	1.93	1.67	2055	1018	1075	803
703	792	270	9	29	122	14.856	0.1723	0.2	2.45	1.64	2012	997	1062	792
703	791	270	9	29	123	13.351	0.1774	1.0	2.21	1.66	2017	999	1062	791
703	787	270	9	30	124	9.676	0.1719	-1.6	1.60	1.69	2076	1028	1053	787
704	1008	271	12	22	92	13.690	0.2172	-0.4	1.70	1.60	2055	1018	1372	1008
704	970	271	14	19	79	12.984	0.2471	-1.0	1.38	1.57	1958	970	1473	1064
704	932	271	12	22	93	11.446	0.2224	0.6	1.43	1.62	1919	950	1270	932
704	893	271	10	26	111	13.342	0.1926	1.6	1.99	1.64	2084	1032	1203	893
704	816	271	10	27	112	9.171	0.1920	-1.2	1.37	1.68	1984	983	1099	816
704	814	271	9	29	124	12.045	0.1776	1.8	2.00	1.67	2082	1031	1091	814
704	802	271	9	30	124	10.142	0.1719	-1.6	1.69	1.69	2111	1045	1074	802
704	788	271	9	29	123	13.690	0.1774	1.0	2.27	1.65	2008	994	1057	788
705	1019	271	14	19	80	10.586	0.2523	0.3	1.13	1.59	2057	1019	1528	1103
705	1018	271	12	22	92	14.020	0.2172	-0.4	1.74	1.59	2075	1028	1386	1018
705	966	271	14	19	79	13.332	0.2471	-1.0	1.42	1.57	1950	966	1469	1061
705	954	271	12	22	94	9.165	0.2276	1.7	1.14	1.63	1995	988	1301	954
705	946	271	12	22	93	11.840	0.2224	0.6	1.47	1.61	1946	964	1290	946
705	890	271	10	26	111	13.680	0.1926	1.6	2.04	1.63	2076	1028	1199	890
705	833	271	10	27	112	9.662	0.1870	-1.2	1.45	1.67	2017	999	1120	833
705	813	271	9	30	124	10.586	0.1720	-1.6	1.76	1.69	2136	1058	1089	813
705	807	271	9	29	124	12.420	0.1776	1.8	2.06	1.67	2063	1022	1082	807

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
705	784	271	9	29	123	14.020	0.1724	1.0	2.32	1.65	1995	988	1050	784
705	751	271	9	30	125	8.365	0.1771	-0.8	1.39	1.70	2004	993	1007	751
706	1032	272	12	22	92	14.341	0.2172	-0.4	1.79	1.59	2103	1041	1405	1032
706	1010	272	14	19	80	11.012	0.2523	0.3	1.18	1.58	2039	1010	1518	1096
706	975	272	14	19	79	13.670	0.2421	-1.0	1.46	1.56	1968	975	1483	1075
706	963	272	12	22	93	12.221	0.2224	0.6	1.53	1.61	1979	980	1313	963
706	944	272	12	22	94	9.655	0.2276	1.7	1.21	1.63	1969	975	1288	944
706	886	272	10	26	111	14.010	0.1926	1.6	2.10	1.63	2067	1024	1194	886
706	851	272	10	27	112	10.127	0.1870	-1.2	1.52	1.67	2056	1018	1144	851
706	826	272	9	30	124	11.012	0.1720	-1.6	1.84	1.68	2167	1073	1106	826
706	802	272	9	29	124	12.784	0.1776	1.8	2.13	1.66	2049	1015	1075	802
706	788	272	9	29	123	14.341	0.1724	1.0	2.38	1.65	2005	993	1056	788
706	770	272	9	30	125	8.900	0.1771	-0.8	1.49	1.70	2048	1014	1033	770
707	1041	272	12	22	92	14.655	0.2172	-0.4	1.83	1.59	2121	1051	1418	1041
707	1001	272	14	19	80	11.421	0.2523	0.3	1.22	1.58	2022	1001	1508	1088
707	986	272	14	19	79	14.000	0.2421	-1.0	1.50	1.56	1991	986	1501	1088
707	976	272	12	22	93	12.590	0.2224	0.6	1.57	1.61	2003	992	1331	976
707	935	272	12	22	94	10.120	0.2276	1.7	1.27	1.63	1944	963	1276	935
707	882	272	10	26	111	14.331	0.1926	1.6	2.14	1.63	2056	1018	1189	882
707	865	272	10	27	112	10.571	0.1870	-1.2	1.59	1.67	2084	1032	1163	865
707	836	272	9	30	124	11.421	0.1720	-1.6	1.90	1.68	2189	1084	1120	836
707	797	272	9	29	124	13.136	0.1776	1.8	2.19	1.66	2035	1008	1069	797
707	795	272	9	29	123	14.655	0.1724	1.0	2.43	1.64	2021	1001	1064	795
707	785	272	9	30	125	9.403	0.1771	-0.8	1.57	1.70	2080	1030	1052	785
708	1051	272	12	22	92	14.961	0.2122	-0.4	1.86	1.59	2137	1058	1428	1051
708	997	272	14	19	79	14.321	0.2421	-1.0	1.53	1.56	2014	997	1519	1100
708	994	272	14	19	80	11.815	0.2523	0.3	1.27	1.58	2006	994	1499	1082
708	990	272	12	22	93	12.947	0.2174	0.6	1.62	1.60	2023	1002	1346	990
708	927	272	12	22	94	10.564	0.2276	1.7	1.32	1.62	1924	953	1266	927
708	878	272	10	26	111	14.644	0.1876	1.6	2.19	1.62	2044	1012	1181	878
708	877	272	10	27	112	10.997	0.1870	-1.2	1.65	1.66	2110	1045	1179	877
708	845	272	9	30	124	11.815	0.1719	-1.6	1.97	1.68	2210	1094	1132	845
708	801	272	9	29	123	14.961	0.1724	1.0	2.48	1.64	2037	1009	1073	801
708	799	272	9	30	125	9.880	0.1721	-0.8	1.65	1.69	2107	1044	1069	799
708	792	272	9	29	124	13.478	0.1776	1.8	2.24	1.66	2022	1001	1062	792
709	1012	273	14	19	79	14.634	0.2421	-1.0	1.57	1.56	2044	1012	1542	1117
709	1005	273	12	22	93	13.294	0.2174	0.6	1.67	1.60	2053	1017	1367	1005
709	988	273	14	19	80	12.195	0.2473	0.3	1.31	1.58	1994	988	1492	1080
709	923	273	12	22	94	10.989	0.2226	1.7	1.38	1.62	1908	945	1257	923
709	891	273	10	27	112	11.405	0.1870	-1.2	1.72	1.66	2142	1061	1199	891
709	876	273	10	26	111	14.950	0.1876	1.6	2.24	1.62	2038	1009	1178	876
709	846	273	9	30	124	12.195	0.1720	-1.6	2.04	1.67	2211	1095	1134	846
709	813	273	9	30	125	10.334	0.1721	-0.8	1.73	1.69	2141	1060	1088	813
709	789	273	9	29	124	13.812	0.1776	1.8	2.31	1.65	2013	997	1058	789
709	747	273	9	30	126	8.058	0.1773	0.0	1.35	1.71	2001	991	1001	747
710	1023	273	14	19	79	14.940	0.2421	-1.0	1.60	1.55	2064	1023	1558	1129
710	1015	273	12	22	93	13.632	0.2174	0.6	1.71	1.60	2074	1027	1381	1015
710	982	273	14	19	80	12.563	0.2473	0.3	1.35	1.57	1983	982	1485	1075
710	939	273	12	22	94	11.397	0.2226	1.7	1.43	1.62	1937	959	1278	939
710	902	273	10	27	112	11.798	0.1870	-1.2	1.78	1.66	2165	1072	1213	902
710	839	273	9	30	124	12.563	0.1720	-1.6	2.10	1.67	2192	1086	1126	839
710	824	273	9	30	125	10.767	0.1721	-0.8	1.80	1.69	2165	1072	1103	824
710	786	273	9	29	124	14.136	0.1776	1.8	2.36	1.65	2007	994	1055	786
710	765	273	9	30	126	8.609	0.1773	0.0	1.45	1.70	2040	1010	1025	765
711	1025	273	12	22	93	13.960	0.2174	0.6	1.75	1.60	2094	1037	1395	1025
711	977	273	14	19	80	12.920	0.2473	0.3	1.39	1.57	1973	977	1480	1070
711	953	273	12	22	94	11.790	0.2226	1.7	1.48	1.61	1964	973	1299	953
711	912	273	10	27	112	12.178	0.1870	-1.2	1.83	1.65	2187	1083	1227	912
711	834	273	9	30	124	12.920	0.1670	-1.6	2.16	1.67	2173	1076	1116	834
711	834	273	9	30	125	11.183	0.1721	-0.8	1.87	1.68	2188	1084	1117	834
711	794	273	9	29	124	14.453	0.1726	1.8	2.41	1.65	2022	1002	1064	794
711	780	273	9	30	126	9.126	0.1773	0.0	1.53	1.70	2074	1027	1046	780
712	1039	274	12	22	93	14.280	0.2174	0.6	1.79	1.59	2121	1051	1415	1039
712	1028	274	14	19	81	10.534	0.2525	1.5	1.14	1.59	2076	1028	1537	1111
712	975	274	14	19	80	13.266	0.2473	0.3	1.43	1.57	1968	975	1477	1068

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
712	970	274	12	22	94	12.170	0.2226	1.7	1.53	1.61	1997	989	1322	970
712	924	274	10	27	112	12.546	0.1870	-1.2	1.89	1.65	2216	1097	1244	924
712	846	274	9	30	125	11.583	0.1721	-0.8	1.95	1.68	2217	1098	1133	846
712	830	274	9	30	124	13.266	0.1670	-1.6	2.22	1.66	2162	1071	1111	830
712	825	274	10	27	114	8.041	0.1924	0.5	1.22	1.68	2029	1005	1109	825
712	803	274	9	29	124	14.762	0.1726	1.8	2.47	1.64	2046	1014	1076	803
712	797	274	9	30	126	9.614	0.1773	0.0	1.62	1.69	2112	1046	1068	797
713	1048	274	12	22	93	14.593	0.2174	0.6	1.83	1.59	2140	1060	1428	1048
713	1018	274	14	19	81	10.958	0.2525	1.5	1.18	1.59	2055	1018	1525	1102
713	985	274	14	19	80	13.603	0.2473	0.3	1.47	1.56	1989	985	1495	1081
713	983	274	12	22	94	12.537	0.2226	1.7	1.58	1.61	2021	1001	1339	983
713	934	274	10	27	112	12.902	0.1820	-1.2	1.95	1.65	2233	1106	1255	934
713	852	274	9	30	125	11.969	0.1721	-0.8	2.01	1.68	2231	1105	1142	852
713	825	274	9	30	124	13.603	0.1670	-1.6	2.28	1.66	2149	1064	1105	825
713	812	274	10	27	114	8.591	0.1924	0.5	1.30	1.68	1989	985	1092	812
713	809	274	9	30	126	10.077	0.1773	0.0	1.70	1.69	2140	1060	1085	809
714	1061	275	12	22	93	14.898	0.2174	0.6	1.88	1.59	2166	1073	1446	1061
714	1011	275	14	19	81	11.365	0.2525	1.5	1.23	1.58	2040	1011	1517	1096
714	1001	275	14	19	80	13.931	0.2473	0.3	1.51	1.56	2021	1001	1520	1099
714	999	275	12	22	94	12.893	0.2226	1.7	1.63	1.60	2052	1016	1361	999
714	945	275	10	27	112	13.247	0.1820	-1.2	2.01	1.64	2256	1117	1269	945
714	846	275	9	30	125	12.342	0.1721	-0.8	2.08	1.67	2214	1096	1134	846
714	827	275	10	27	114	9.107	0.1924	0.5	1.39	1.68	2019	1000	1112	827
714	824	275	9	30	126	10.519	0.1723	0.0	1.78	1.69	2171	1075	1103	824
714	822	275	9	30	124	13.931	0.1670	-1.6	2.34	1.66	2140	1060	1101	822
715	1011	275	14	19	80	14.250	0.2423	0.3	1.54	1.56	2042	1011	1535	1113
715	1010	275	12	22	94	13.238	0.2226	1.7	1.67	1.60	2074	1027	1378	1010
715	1003	275	14	19	81	11.757	0.2525	1.5	1.27	1.58	2025	1003	1509	1090
715	939	275	10	27	112	13.584	0.1820	-1.2	2.06	1.64	2241	1110	1261	939
715	844	275	10	27	114	9.594	0.1924	0.5	1.46	1.67	2054	1017	1135	844
715	840	275	9	30	125	12.703	0.1721	-0.8	2.14	1.67	2196	1088	1126	840
715	834	275	9	30	126	10.943	0.1723	0.0	1.85	1.69	2194	1087	1117	834
715	829	275	9	30	124	14.250	0.1670	-1.6	2.39	1.65	2158	1069	1110	829
716	1023	275	12	22	94	13.574	0.2176	1.7	1.71	1.60	2092	1036	1390	1023
716	1022	275	14	19	80	14.562	0.2423	0.3	1.57	1.56	2063	1022	1552	1126
716	996	275	14	19	81	12.135	0.2525	1.5	1.31	1.58	2012	996	1501	1084
716	933	275	10	27	112	13.911	0.1820	-1.2	2.10	1.64	2228	1103	1254	933
716	858	275	10	27	114	10.056	0.1924	0.5	1.53	1.67	2086	1033	1155	858
716	844	275	9	30	126	11.349	0.1723	0.0	1.91	1.68	2216	1098	1130	844
716	835	275	9	30	124	14.562	0.1670	-1.6	2.45	1.65	2175	1077	1119	835
716	834	275	9	30	125	13.053	0.1721	-0.8	2.20	1.67	2180	1080	1119	834
717	1036	276	14	19	80	14.866	0.2423	0.3	1.61	1.55	2093	1036	1575	1142
717	1036	276	12	22	94	13.901	0.2176	1.7	1.76	1.60	2120	1050	1409	1036
717	1017	276	12	23	95	9.088	0.2169	-1.7	1.16	1.64	2184	1082	1383	1017
717	993	276	14	19	81	12.502	0.2525	1.5	1.36	1.57	2004	993	1497	1081
717	929	276	10	27	112	14.230	0.1820	-1.2	2.16	1.64	2219	1099	1250	929
717	876	276	10	27	114	10.497	0.1874	0.5	1.60	1.67	2120	1050	1177	876
717	856	276	9	30	126	11.740	0.1723	0.0	1.99	1.68	2245	1112	1146	856
717	845	276	9	30	124	14.866	0.1670	-1.6	2.50	1.65	2200	1090	1132	845
717	831	276	9	30	125	13.393	0.1671	-0.8	2.26	1.66	2167	1073	1112	831
717	829	276	10	27	115	8.013	0.1925	1.4	1.22	1.68	2041	1011	1113	829
718	1046	276	12	22	94	14.220	0.2176	1.7	1.80	1.59	2140	1060	1422	1046
718	1005	276	12	23	95	9.574	0.2169	-1.7	1.22	1.64	2151	1065	1367	1005
718	986	276	14	19	81	12.857	0.2475	1.5	1.40	1.57	1991	986	1489	1078
718	925	276	10	27	112	14.541	0.1820	-1.2	2.21	1.63	2208	1094	1244	925
718	889	276	10	27	114	10.920	0.1874	0.5	1.66	1.67	2146	1063	1194	889
718	852	276	9	30	126	12.118	0.1723	0.0	2.05	1.68	2233	1106	1141	852
718	826	276	9	30	125	13.724	0.1671	-0.8	2.32	1.66	2154	1067	1106	826
718	815	276	10	27	115	8.561	0.1925	1.4	1.31	1.68	2001	991	1096	815
718	755	276	9	30	128	8.007	0.1776	1.6	1.36	1.71	2028	1004	1011	755
719	1059	277	12	22	94	14.531	0.2176	1.7	1.84	1.59	2166	1073	1441	1059
719	996	277	12	23	95	10.035	0.2169	-1.7	1.28	1.64	2127	1054	1356	996
719	987	277	14	19	81	13.201	0.2475	1.5	1.44	1.57	1993	987	1492	1080
719	922	277	10	27	112	14.845	0.1820	-1.2	2.26	1.63	2202	1091	1241	922
719	903	277	10	27	114	11.325	0.1874	0.5	1.73	1.66	2178	1079	1214	903

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
719	846	277	9	30	126	12.484	0.1723	0.0	2.12	1.67	2217	1098	1134	846
719	833	277	10	27	115	9.075	0.1925	1.4	1.39	1.68	2037	1009	1120	833
719	831	277	9	30	125	14.047	0.1671	-0.8	2.38	1.66	2166	1073	1112	831
719	776	277	9	30	128	8.555	0.1776	1.6	1.46	1.70	2075	1028	1038	776
720	1068	277	12	22	94	14.835	0.2176	1.7	1.88	1.59	2185	1082	1453	1068
720	999	277	14	19	81	13.536	0.2475	1.5	1.47	1.57	2017	999	1511	1094
720	989	277	12	23	95	10.475	0.2119	-1.7	1.34	1.63	2101	1041	1342	989
720	914	277	10	27	114	11.716	0.1874	0.5	1.79	1.66	2201	1090	1229	914
720	850	277	10	27	115	9.560	0.1925	1.4	1.46	1.67	2072	1026	1142	850
720	840	277	9	30	126	12.839	0.1723	0.0	2.18	1.67	2200	1090	1126	840
720	838	277	9	30	125	14.362	0.1671	-0.8	2.43	1.65	2184	1082	1122	838
720	791	277	9	30	128	9.069	0.1776	1.6	1.54	1.70	2109	1045	1059	791
721	1011	277	14	19	81	13.862	0.2475	1.5	1.51	1.56	2041	1011	1530	1107
721	991	277	12	23	95	10.897	0.2119	-1.7	1.39	1.63	2103	1041	1345	991
721	924	277	10	27	114	12.093	0.1874	0.5	1.85	1.66	2223	1101	1243	924
721	864	277	10	27	115	10.021	0.1925	1.4	1.53	1.67	2103	1042	1163	864
721	845	277	9	30	125	14.669	0.1671	-0.8	2.48	1.65	2202	1090	1131	845
721	835	277	9	30	126	13.183	0.1723	0.0	2.23	1.66	2185	1082	1119	835
721	805	277	9	30	128	9.554	0.1776	1.6	1.63	1.70	2140	1060	1078	805
722	1026	278	14	19	81	14.180	0.2475	1.5	1.55	1.56	2072	1026	1554	1124
722	1011	278	12	23	95	11.302	0.2119	-1.7	1.45	1.63	2142	1061	1373	1011
722	937	278	10	27	114	12.458	0.1874	0.5	1.91	1.65	2252	1115	1260	937
722	881	278	10	27	115	10.461	0.1925	1.4	1.61	1.67	2140	1060	1186	881
722	854	278	9	30	125	14.969	0.1671	-0.8	2.54	1.65	2227	1103	1144	854
722	831	278	9	30	126	13.517	0.1723	0.0	2.30	1.66	2175	1077	1114	831
722	820	278	9	30	128	10.014	0.1776	1.6	1.71	1.69	2175	1077	1099	820
723	1037	278	14	19	81	14.491	0.2475	1.5	1.58	1.56	2093	1037	1571	1136
723	1026	278	12	23	95	11.691	0.2119	-1.7	1.49	1.62	2171	1075	1393	1026
723	1022	278	12	23	96	9.050	0.2171	-0.6	1.16	1.64	2198	1089	1389	1022
723	946	278	10	27	114	12.812	0.1874	0.5	1.96	1.65	2272	1125	1272	946
723	893	278	10	27	115	10.882	0.1925	1.4	1.67	1.66	2166	1073	1203	893
723	831	278	9	30	128	10.453	0.1776	1.6	1.78	1.69	2200	1090	1114	831
723	830	278	9	30	126	13.843	0.1673	0.0	2.35	1.66	2167	1073	1111	830
723	801	278	9	31	128	8.260	0.1719	-1.7	1.41	1.71	2191	1085	1072	801
724	1046	278	14	19	81	14.794	0.2425	1.5	1.61	1.55	2112	1046	1584	1150
724	1040	278	12	23	95	12.068	0.2119	-1.7	1.54	1.62	2198	1089	1413	1040
724	1010	278	12	23	96	9.534	0.2171	-0.6	1.22	1.64	2166	1073	1373	1010
724	950	278	10	27	114	13.155	0.1874	0.5	2.01	1.65	2282	1130	1279	950
724	906	278	10	27	115	11.286	0.1875	1.4	1.73	1.66	2189	1084	1217	906
724	841	278	9	30	128	10.874	0.1776	1.6	1.85	1.69	2224	1102	1128	841
724	837	278	9	30	126	14.161	0.1673	0.0	2.41	1.66	2185	1082	1121	837
724	817	278	9	31	128	8.788	0.1719	-1.7	1.50	1.71	2230	1104	1095	817
725	1057	279	12	23	95	12.432	0.2119	-1.7	1.59	1.62	2232	1105	1437	1057
725	1001	279	12	23	96	9.993	0.2171	-0.6	1.28	1.64	2142	1061	1361	1001
725	945	279	10	27	114	13.489	0.1874	0.5	2.07	1.64	2270	1124	1273	945
725	920	279	10	27	115	11.675	0.1875	1.4	1.80	1.66	2220	1099	1236	920
726	1069	279	12	23	95	12.785	0.2119	-1.7	1.64	1.61	2256	1117	1453	1069
726	992	279	12	23	96	10.432	0.2171	-0.6	1.34	1.63	2119	1049	1350	992
726	941	279	10	27	114	13.814	0.1824	0.5	2.12	1.64	2253	1116	1264	941
726	930	279	10	27	115	12.051	0.1875	1.4	1.85	1.66	2241	1110	1250	930
727	1087	280	12	23	95	13.128	0.2069	-1.7	1.69	1.61	2285	1132	1473	1087
727	1002	280	12	23	96	10.852	0.2121	-0.6	1.40	1.63	2130	1055	1360	1002
727	943	280	10	27	115	12.415	0.1875	1.4	1.92	1.65	2270	1124	1267	943
727	937	280	10	27	114	14.131	0.1824	0.5	2.18	1.64	2244	1112	1259	937
728	1098	280	12	23	95	13.461	0.2069	-1.7	1.73	1.61	2307	1143	1488	1098
728	1018	280	12	23	96	11.255	0.2121	-0.6	1.45	1.63	2161	1071	1382	1018
728	952	280	10	27	115	12.768	0.1875	1.4	1.97	1.65	2290	1134	1280	952
728	933	280	10	27	114	14.441	0.1824	0.5	2.22	1.63	2234	1106	1253	933
728	860	280	10	28	116	8.502	0.1870	-1.4	1.32	1.69	2163	1071	1156	860
729	1108	280	12	23	95	13.786	0.2069	-1.7	1.77	1.61	2329	1153	1503	1108
729	1033	280	12	23	96	11.643	0.2121	-0.6	1.50	1.63	2190	1085	1403	1033
729	1027	280	12	23	97	9.012	0.2173	0.4	1.16	1.64	2213	1096	1394	1027
729	954	280	10	27	115	13.110	0.1875	1.4	2.02	1.65	2295	1137	1283	954
729	929	280	10	27	114	14.743	0.1824	0.5	2.27	1.63	2224	1102	1248	929
729	881	280	10	28	116	9.012	0.1820	-1.4	1.40	1.69	2203	1091	1181	881

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
730	1123	281	12	23	95	14.102	0.2069	-1.7	1.82	1.60	2358	1168	1523	1123
730	1051	281	12	23	96	12.018	0.2121	-0.6	1.55	1.62	2226	1102	1428	1051
730	1016	281	12	23	97	9.495	0.2173	0.4	1.23	1.64	2183	1081	1381	1016
730	949	281	10	27	115	13.443	0.1875	1.4	2.08	1.64	2282	1130	1277	949
730	901	281	10	28	116	9.495	0.1820	-1.4	1.48	1.68	2248	1113	1209	901
731	1132	281	12	23	95	14.411	0.2069	-1.7	1.86	1.60	2378	1178	1537	1132
731	1064	281	12	23	96	12.381	0.2121	-0.6	1.60	1.62	2251	1115	1446	1064
731	1006	281	12	23	97	9.952	0.2173	0.4	1.29	1.64	2156	1068	1366	1006
731	944	281	10	27	115	13.767	0.1875	1.4	2.13	1.64	2269	1124	1270	944
731	916	281	10	28	116	9.952	0.1820	-1.4	1.55	1.68	2280	1129	1230	916
732	1163	282	14	20	83	9.945	0.2420	-1.2	1.11	1.60	2347	1163	1674	1215
732	1146	282	12	23	95	14.712	0.2069	-1.7	1.90	1.60	2406	1192	1555	1146
732	1080	282	12	23	96	12.732	0.2121	-0.6	1.65	1.62	2284	1131	1468	1080
732	999	282	12	23	97	10.389	0.2173	0.4	1.35	1.63	2136	1058	1357	999
732	941	282	10	27	115	14.083	0.1825	1.4	2.18	1.64	2257	1118	1264	941
732	934	282	10	28	116	10.389	0.1820	-1.4	1.62	1.68	2319	1148	1254	934
732	891	282	9	31	128	12.188	0.1669	-1.7	2.11	1.68	2393	1185	1194	891
732	876	282	9	31	129	10.389	0.1671	-0.9	1.80	1.70	2367	1172	1171	876
732	841	282	9	30	128	13.757	0.1726	1.6	2.37	1.66	2205	1092	1127	841
732	812	282	9	31	130	8.209	0.1722	-0.2	1.42	1.71	2228	1103	1086	812
733	1149	282	14	20	83	10.382	0.2420	-1.2	1.16	1.60	2319	1149	1658	1203
733	1092	282	12	23	96	13.074	0.2121	-0.6	1.69	1.61	2307	1143	1485	1092
733	1008	282	12	23	97	10.807	0.2173	0.4	1.40	1.63	2152	1066	1370	1008
733	946	282	10	28	116	10.807	0.1820	-1.4	1.68	1.67	2346	1162	1271	946
733	937	282	10	27	115	14.391	0.1825	1.4	2.23	1.63	2246	1113	1258	937
733	886	282	9	31	129	10.807	0.1671	-0.9	1.87	1.69	2392	1185	1185	886
733	885	282	9	31	128	12.545	0.1669	-1.7	2.17	1.68	2373	1175	1185	885
733	866	282	10	28	117	8.473	0.1872	-0.5	1.32	1.69	2182	1081	1164	866
733	848	282	9	30	128	14.073	0.1726	1.6	2.43	1.66	2223	1101	1136	848
733	828	282	9	31	130	8.734	0.1722	-0.2	1.51	1.71	2266	1122	1109	828
734	1137	282	14	20	83	10.800	0.2420	-1.2	1.20	1.60	2296	1137	1645	1193
734	1105	282	12	23	96	13.406	0.2071	-0.6	1.73	1.61	2327	1153	1498	1105
734	1045	282	12	23	98	8.467	0.2225	1.4	1.10	1.65	2272	1125	1423	1045
734	1024	282	12	23	97	11.208	0.2173	0.4	1.45	1.63	2183	1081	1392	1024
734	958	282	10	28	116	11.208	0.1820	-1.4	1.74	1.67	2372	1175	1287	958
734	933	282	10	27	115	14.692	0.1825	1.4	2.28	1.63	2236	1108	1253	933
734	896	282	9	31	129	11.208	0.1671	-0.9	1.94	1.69	2415	1196	1199	896
734	886	282	10	28	117	8.982	0.1872	-0.5	1.40	1.69	2223	1101	1190	886
734	879	282	9	31	128	12.891	0.1669	-1.7	2.23	1.67	2356	1167	1177	879
734	854	282	9	30	128	14.381	0.1726	1.6	2.48	1.65	2240	1110	1145	854
734	843	282	9	31	130	9.228	0.1722	-0.2	1.60	1.70	2300	1139	1129	843
735	1127	283	14	20	83	11.201	0.2370	-1.2	1.25	1.60	2276	1127	1633	1188
735	1120	283	12	23	96	13.729	0.2071	-0.6	1.78	1.61	2357	1168	1518	1120
735	1045	283	12	23	97	11.595	0.2123	0.4	1.51	1.63	2218	1099	1417	1045
735	1033	283	12	23	98	8.976	0.2175	1.4	1.17	1.64	2231	1105	1402	1033
735	973	283	10	28	116	11.595	0.1820	-1.4	1.81	1.67	2405	1191	1307	973
735	930	283	10	27	115	14.987	0.1825	1.4	2.33	1.63	2231	1105	1250	930
735	906	283	10	28	117	9.462	0.1872	-0.5	1.48	1.68	2268	1124	1218	906
735	905	283	9	31	129	11.595	0.1671	-0.9	2.01	1.69	2436	1207	1211	905
735	874	283	9	31	128	13.227	0.1669	-1.7	2.29	1.67	2344	1161	1172	874
735	865	283	9	30	128	14.682	0.1676	1.6	2.54	1.65	2264	1121	1157	865
735	860	283	9	31	130	9.696	0.1722	-0.2	1.69	1.70	2339	1158	1151	860
736	1130	283	12	23	96	14.044	0.2071	-0.6	1.82	1.60	2378	1178	1533	1130
736	1119	283	14	20	83	11.587	0.2370	-1.2	1.29	1.59	2259	1119	1624	1181
736	1059	283	12	23	97	11.969	0.2123	0.4	1.56	1.62	2245	1112	1436	1059
736	1021	283	12	23	98	9.456	0.2175	1.4	1.23	1.64	2198	1089	1386	1021
736	983	283	10	28	116	11.969	0.1820	-1.4	1.87	1.66	2428	1203	1321	983
736	923	283	10	28	117	9.918	0.1822	-0.5	1.55	1.68	2299	1139	1237	923
736	897	283	9	31	129	11.969	0.1671	-0.9	2.08	1.68	2413	1195	1201	897
736	874	283	9	31	128	13.555	0.1619	-1.7	2.35	1.67	2336	1157	1169	874
736	872	283	9	31	130	10.142	0.1722	-0.2	1.76	1.70	2366	1172	1168	872
736	871	283	9	30	128	14.976	0.1676	1.6	2.59	1.65	2280	1130	1166	871
737	1140	283	12	23	96	14.352	0.2071	-0.6	1.86	1.60	2398	1188	1546	1140
737	1111	283	14	20	83	11.961	0.2370	-1.2	1.33	1.59	2244	1111	1616	1175
737	1072	283	12	23	97	12.330	0.2123	0.4	1.60	1.62	2271	1125	1454	1072

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
737	1011	283	12	23	98	9.912	0.2175	1.4	1.29	1.64	2170	1075	1372	1011
737	992	283	10	28	116	12.330	0.1820	-1.4	1.92	1.66	2450	1214	1334	992
737	937	283	10	28	117	10.353	0.1822	-0.5	1.62	1.68	2329	1154	1257	937
737	890	283	9	31	129	12.330	0.1671	-0.9	2.14	1.68	2392	1185	1192	890
737	883	283	9	31	130	10.568	0.1722	-0.2	1.84	1.69	2392	1185	1183	883
737	881	283	9	31	128	13.873	0.1619	-1.7	2.40	1.66	2356	1167	1179	881
738	1153	284	12	23	96	14.652	0.2071	-0.6	1.91	1.60	2426	1202	1564	1153
738	1107	284	14	20	83	12.322	0.2370	-1.2	1.38	1.59	2234	1107	1611	1171
738	1088	284	12	23	97	12.680	0.2123	0.4	1.65	1.62	2304	1141	1477	1088
738	1006	284	10	28	116	12.680	0.1770	-1.4	1.98	1.66	2477	1227	1350	1006
738	1004	284	12	23	98	10.346	0.2175	1.4	1.35	1.63	2150	1065	1363	1004
738	953	284	10	28	117	10.770	0.1822	-0.5	1.69	1.67	2365	1172	1279	953
738	897	284	9	31	130	10.976	0.1672	-0.2	1.91	1.69	2422	1200	1199	897
738	891	284	9	31	128	14.184	0.1619	-1.7	2.46	1.66	2383	1180	1192	891
738	885	284	9	31	129	12.680	0.1671	-0.9	2.20	1.68	2377	1177	1185	885
738	872	284	10	28	118	8.444	0.1873	0.3	1.33	1.69	2200	1090	1172	872
739	1171	284	14	20	84	9.898	0.2423	0.0	1.11	1.60	2365	1171	1681	1221
739	1162	284	12	23	96	14.946	0.2071	-0.6	1.94	1.60	2445	1211	1577	1162
739	1102	284	14	20	83	12.672	0.2370	-1.2	1.42	1.58	2224	1102	1606	1167
739	1100	284	12	23	97	13.020	0.2123	0.4	1.70	1.61	2327	1153	1493	1100
739	1015	284	12	23	98	10.763	0.2175	1.4	1.41	1.63	2171	1075	1379	1015
739	1004	284	10	28	116	13.020	0.1770	-1.4	2.04	1.65	2470	1223	1347	1004
739	965	284	10	28	117	11.170	0.1822	-0.5	1.75	1.67	2391	1184	1295	965
739	907	284	9	31	130	11.369	0.1672	-0.2	1.98	1.69	2445	1211	1213	907
739	898	284	9	31	128	14.488	0.1619	-1.7	2.51	1.66	2401	1189	1202	898
739	891	284	10	28	118	8.951	0.1873	0.3	1.41	1.69	2242	1111	1198	891
739	879	284	9	31	129	13.020	0.1671	-0.9	2.26	1.67	2361	1169	1178	879
740	1159	285	14	20	84	10.332	0.2423	0.0	1.16	1.60	2340	1159	1668	1211
740	1120	285	14	20	83	13.012	0.2370	-1.2	1.46	1.58	2261	1120	1634	1187
740	1115	285	12	23	97	13.351	0.2123	0.4	1.75	1.61	2358	1168	1515	1115
740	1035	285	12	23	98	11.163	0.2175	1.4	1.46	1.63	2211	1095	1407	1035
740	999	285	10	28	116	13.351	0.1770	-1.4	2.09	1.65	2457	1217	1341	999
740	979	285	10	28	117	11.556	0.1822	-0.5	1.82	1.67	2424	1201	1315	979
740	912	285	10	28	118	9.430	0.1873	0.3	1.49	1.68	2287	1133	1226	912
740	908	285	9	31	128	14.785	0.1619	-1.7	2.57	1.65	2428	1202	1215	908
740	905	285	9	31	130	11.748	0.1672	-0.2	2.05	1.68	2437	1207	1210	905
740	876	285	9	31	129	13.351	0.1671	-0.9	2.33	1.67	2349	1164	1173	876
741	1147	285	14	20	84	10.748	0.2423	0.0	1.21	1.60	2316	1147	1655	1201
741	1133	285	14	20	83	13.342	0.2370	-1.2	1.50	1.58	2287	1133	1654	1201
741	1126	285	12	23	97	13.673	0.2123	0.4	1.79	1.61	2380	1179	1529	1126
741	1050	285	12	23	98	11.548	0.2175	1.4	1.51	1.63	2240	1109	1428	1050
741	993	285	10	28	116	13.673	0.1770	-1.4	2.14	1.65	2442	1209	1334	993
741	989	285	10	28	117	11.928	0.1822	-0.5	1.88	1.66	2447	1212	1329	989
741	927	285	10	28	118	9.885	0.1873	0.3	1.56	1.68	2320	1149	1247	927
741	897	285	9	31	130	12.114	0.1672	-0.2	2.12	1.68	2415	1196	1201	897
741	882	285	9	31	129	13.673	0.1671	-0.9	2.38	1.67	2367	1172	1182	882
741	819	285	9	31	132	8.159	0.1725	1.4	1.43	1.71	2256	1117	1095	819
742	1144	285	14	20	83	13.664	0.2320	-1.2	1.53	1.58	2310	1144	1672	1218
742	1138	285	12	23	97	13.987	0.2073	0.4	1.83	1.61	2398	1188	1541	1138
742	1137	285	14	20	84	11.148	0.2423	0.0	1.25	1.60	2296	1137	1644	1193
742	1066	285	12	23	98	11.920	0.2125	1.4	1.56	1.62	2265	1122	1446	1066
742	999	285	10	28	117	12.288	0.1822	-0.5	1.93	1.66	2469	1223	1342	999
742	988	285	10	28	116	13.987	0.1770	-1.4	2.19	1.64	2429	1203	1327	988
742	943	285	10	28	118	10.318	0.1823	0.3	1.62	1.68	2348	1163	1265	943
742	890	285	9	31	130	12.468	0.1672	-0.2	2.18	1.68	2395	1186	1192	890
742	890	285	9	31	129	13.987	0.1621	-0.9	2.44	1.66	2384	1181	1191	890
742	837	285	9	31	132	8.681	0.1725	1.4	1.52	1.71	2295	1137	1118	837
743	1161	286	14	20	83	13.978	0.2320	-1.2	1.57	1.57	2344	1161	1697	1236
743	1152	286	12	23	97	14.293	0.2073	0.4	1.87	1.60	2426	1202	1561	1152
743	1130	286	14	20	84	11.532	0.2423	0.0	1.30	1.59	2282	1130	1636	1187
743	1083	286	12	23	98	12.280	0.2125	1.4	1.61	1.62	2299	1139	1469	1083
743	1012	286	10	28	117	12.637	0.1822	-0.5	1.99	1.66	2499	1238	1360	1012
743	985	286	10	28	116	14.293	0.1770	-1.4	2.25	1.64	2420	1199	1323	985
743	959	286	10	28	118	10.734	0.1823	0.3	1.70	1.67	2384	1181	1287	959
743	901	286	9	31	129	14.293	0.1621	-0.9	2.50	1.66	2411	1194	1205	901

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
743	886	286	9	31	130	12.813	0.1672	-0.2	2.24	1.67	2381	1179	1186	886
743	878	286	10	28	119	8.415	0.1875	1.2	1.33	1.69	2219	1099	1179	878
743	854	286	9	31	132	9.172	0.1725	1.4	1.61	1.70	2337	1157	1143	854
744	1172	286	14	20	83	14.284	0.2320	-1.2	1.60	1.57	2367	1172	1715	1248
744	1161	286	12	23	97	14.593	0.2073	0.4	1.91	1.60	2446	1212	1574	1161
744	1121	286	14	20	84	11.904	0.2373	0.0	1.34	1.59	2264	1121	1624	1183
744	1096	286	12	23	98	12.629	0.2125	1.4	1.66	1.62	2324	1151	1486	1096
744	1008	286	10	28	117	12.976	0.1772	-0.5	2.04	1.65	2484	1230	1352	1008
744	980	286	10	28	116	14.593	0.1770	-1.4	2.29	1.64	2409	1193	1317	980
744	971	286	10	28	118	11.133	0.1823	0.3	1.76	1.67	2410	1194	1303	971
744	907	286	9	31	129	14.593	0.1621	-0.9	2.55	1.66	2429	1203	1213	907
744	898	286	10	28	119	8.921	0.1875	1.2	1.41	1.69	2261	1120	1205	898
744	880	286	9	31	130	13.147	0.1672	-0.2	2.30	1.67	2366	1172	1178	880
744	867	286	9	31	132	9.637	0.1725	1.4	1.69	1.70	2367	1173	1160	867
745	1188	287	14	20	83	14.583	0.2320	-1.2	1.64	1.57	2399	1188	1738	1265
745	1174	287	12	23	97	14.885	0.2073	0.4	1.96	1.60	2474	1225	1592	1174
745	1116	287	14	20	84	12.263	0.2373	0.0	1.39	1.59	2254	1116	1620	1179
745	1112	287	12	23	98	12.968	0.2125	1.4	1.71	1.61	2355	1167	1508	1112
745	1003	287	10	28	117	13.306	0.1772	-0.5	2.10	1.65	2470	1224	1346	1003
745	985	287	10	28	118	11.517	0.1823	0.3	1.82	1.67	2443	1210	1323	985
745	977	287	10	28	116	14.885	0.1770	-1.4	2.35	1.63	2403	1190	1314	977
745	918	287	10	28	119	9.398	0.1875	1.2	1.49	1.68	2306	1142	1233	918
745	916	287	9	31	129	14.885	0.1621	-0.9	2.61	1.65	2455	1216	1226	916
745	884	287	9	31	130	13.472	0.1672	-0.2	2.36	1.67	2376	1177	1184	884
745	882	287	9	31	132	10.080	0.1725	1.4	1.78	1.70	2403	1190	1181	882
746	1199	287	14	20	83	14.875	0.2320	-1.2	1.68	1.57	2421	1199	1755	1278
746	1183	287	14	20	85	9.851	0.2475	1.2	1.12	1.60	2388	1183	1693	1227
746	1122	287	12	23	98	13.297	0.2125	1.4	1.75	1.61	2378	1178	1524	1122
746	1116	287	14	20	84	12.612	0.2373	0.0	1.42	1.59	2254	1116	1622	1180
746	1113	287	12	24	99	8.399	0.2069	-1.8	1.11	1.66	2473	1225	1506	1113
746	998	287	10	28	117	13.627	0.1772	-0.5	2.15	1.65	2455	1216	1338	998
746	996	287	10	28	118	11.888	0.1823	0.3	1.88	1.66	2466	1222	1337	996
746	933	287	10	28	119	9.851	0.1875	1.2	1.56	1.68	2339	1159	1254	933
746	894	287	9	31	132	10.503	0.1725	1.4	1.85	1.69	2429	1203	1196	894
746	891	287	9	31	130	13.789	0.1672	-0.2	2.42	1.66	2395	1186	1194	891
746	865	287	9	32	132	8.399	0.1669	-1.8	1.48	1.72	2429	1203	1157	865
747	1167	287	14	20	85	10.284	0.2425	1.2	1.16	1.60	2357	1167	1675	1217
747	1133	287	12	23	98	13.618	0.2125	1.4	1.79	1.61	2399	1188	1538	1133
747	1130	287	14	20	84	12.950	0.2373	0.0	1.46	1.58	2282	1130	1644	1196
747	1097	287	12	24	99	8.903	0.2069	-1.8	1.18	1.65	2429	1203	1485	1097
747	1005	287	10	28	118	12.247	0.1823	0.3	1.94	1.66	2489	1233	1351	1005
747	992	287	10	28	117	13.940	0.1772	-0.5	2.20	1.65	2442	1210	1332	992
747	947	287	10	28	119	10.284	0.1875	1.2	1.63	1.68	2370	1174	1274	947
747	904	287	9	31	132	10.909	0.1725	1.4	1.92	1.69	2453	1215	1210	904
747	898	287	9	31	130	14.098	0.1672	-0.2	2.47	1.66	2414	1196	1204	898
747	882	287	9	32	132	8.903	0.1669	-1.8	1.57	1.71	2467	1222	1179	882
748	1157	288	14	20	85	10.698	0.2425	1.2	1.22	1.60	2336	1157	1664	1209
748	1148	288	14	20	84	13.279	0.2373	0.0	1.50	1.58	2318	1148	1671	1215
748	1148	288	12	23	98	13.931	0.2125	1.4	1.84	1.61	2429	1203	1559	1148
748	1085	288	12	24	99	9.379	0.2069	-1.8	1.24	1.65	2396	1187	1469	1085
748	1018	288	10	28	118	12.595	0.1823	0.3	2.00	1.66	2518	1247	1368	1018
748	989	288	10	28	117	14.245	0.1772	-0.5	2.26	1.64	2433	1205	1327	989
748	965	288	10	28	119	10.698	0.1825	1.2	1.70	1.67	2404	1191	1294	965
748	916	288	9	31	132	11.300	0.1725	1.4	2.00	1.69	2484	1231	1227	916
748	909	288	9	31	130	14.400	0.1622	-0.2	2.53	1.66	2439	1208	1216	909
748	899	288	9	32	132	9.379	0.1669	-1.8	1.66	1.71	2509	1243	1203	899
749	1160	288	14	20	84	13.600	0.2373	0.0	1.54	1.58	2343	1160	1691	1229
749	1157	288	12	23	98	14.236	0.2125	1.4	1.88	1.60	2449	1213	1572	1157
749	1147	288	14	20	85	11.095	0.2425	1.2	1.26	1.60	2316	1147	1653	1201
749	1074	288	12	24	99	9.832	0.2069	-1.8	1.30	1.65	2365	1172	1454	1074
749	1011	288	10	28	118	12.933	0.1823	0.3	2.05	1.65	2500	1238	1359	1011
749	984	288	10	28	117	14.543	0.1772	-0.5	2.30	1.64	2423	1200	1322	984
749	977	288	10	28	119	11.095	0.1825	1.2	1.76	1.67	2430	1203	1310	977
749	916	288	9	31	130	14.695	0.1622	-0.2	2.58	1.66	2457	1217	1225	916
749	912	288	9	32	132	9.832	0.1669	-1.8	1.74	1.70	2539	1258	1221	912

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
749	910	288	9	31	132	11.677	0.1675	1.4	2.06	1.69	2459	1218	1216	910
750	1173	288	14	20	84	13.912	0.2373	0.0	1.57	1.57	2367	1173	1709	1243
750	1169	288	12	23	98	14.534	0.2075	1.4	1.92	1.60	2466	1221	1583	1169
750	1138	288	14	20	85	11.478	0.2425	1.2	1.30	1.59	2298	1138	1643	1193
750	1067	288	12	24	99	10.263	0.2069	-1.8	1.36	1.64	2346	1162	1446	1067
750	1005	288	10	28	118	13.261	0.1823	0.3	2.10	1.65	2483	1230	1351	1005
750	988	288	10	28	119	11.478	0.1825	1.2	1.82	1.67	2454	1216	1326	988
750	980	288	10	28	117	14.835	0.1772	-0.5	2.35	1.64	2413	1195	1316	980
750	924	288	9	32	132	10.263	0.1669	-1.8	1.81	1.70	2567	1272	1237	924
750	922	288	9	31	130	14.984	0.1622	-0.2	2.63	1.65	2474	1226	1233	922
750	902	288	9	31	132	12.041	0.1675	1.4	2.12	1.68	2437	1207	1207	902
750	859	288	9	32	133	8.110	0.1671	-1.0	1.44	1.72	2419	1198	1149	859
751	1187	289	14	20	84	14.217	0.2323	0.0	1.61	1.57	2397	1187	1731	1262
751	1182	289	12	23	98	14.825	0.2075	1.4	1.96	1.60	2494	1235	1601	1182
751	1133	289	14	20	85	11.848	0.2425	1.2	1.35	1.59	2287	1133	1637	1189
751	1090	289	12	24	99	10.676	0.2069	-1.8	1.42	1.64	2391	1184	1477	1090
751	1002	289	10	28	118	13.581	0.1773	0.3	2.16	1.65	2469	1223	1343	1002
751	1002	289	10	28	119	11.848	0.1825	1.2	1.89	1.66	2486	1231	1344	1002
751	939	289	9	32	132	10.676	0.1619	-1.8	1.89	1.70	2599	1287	1254	939
751	897	289	9	31	132	12.393	0.1675	1.4	2.19	1.68	2420	1199	1199	897
751	880	289	9	32	133	8.629	0.1671	-1.0	1.53	1.71	2468	1222	1176	880
752	1199	289	14	20	84	14.514	0.2323	0.0	1.65	1.57	2420	1199	1748	1274
752	1126	289	14	20	85	12.206	0.2425	1.2	1.39	1.59	2273	1126	1630	1183
752	1117	289	12	24	100	8.365	0.2121	-0.8	1.12	1.66	2493	1235	1515	1117
752	1106	289	12	24	99	11.073	0.2069	-1.8	1.47	1.64	2424	1201	1500	1106
752	1011	289	10	28	119	12.206	0.1825	1.2	1.94	1.66	2508	1242	1358	1011
752	996	289	10	28	118	13.893	0.1773	0.3	2.21	1.65	2455	1216	1336	996
753	1214	290	14	20	84	14.805	0.2323	0.0	1.68	1.57	2451	1214	1772	1291
753	1131	290	14	20	85	12.553	0.2375	1.2	1.43	1.59	2284	1131	1639	1193
753	1126	290	12	24	99	11.455	0.2069	-1.8	1.53	1.64	2464	1220	1527	1126
753	1102	290	12	24	100	8.867	0.2121	-0.8	1.19	1.65	2451	1214	1495	1102
753	1022	290	10	28	119	12.553	0.1825	1.2	2.01	1.66	2533	1255	1373	1022
753	993	290	10	28	118	14.198	0.1773	0.3	2.26	1.64	2447	1212	1332	993
753	929	290	10	29	120	8.359	0.1820	-1.5	1.34	1.70	2404	1191	1247	929
754	1145	290	14	20	85	12.890	0.2375	1.2	1.47	1.58	2312	1145	1661	1209
754	1143	290	12	24	99	11.825	0.2019	-1.8	1.58	1.63	2491	1234	1546	1143
754	1090	290	12	24	100	9.342	0.2071	-0.8	1.25	1.65	2411	1194	1475	1090
754	1015	290	10	28	119	12.890	0.1825	1.2	2.06	1.65	2514	1245	1364	1015
754	988	290	10	28	118	14.495	0.1773	0.3	2.31	1.64	2436	1207	1326	988
754	951	290	10	29	120	8.861	0.1770	-1.5	1.42	1.69	2446	1212	1273	951
755	1158	290	14	20	85	13.217	0.2375	1.2	1.51	1.58	2338	1158	1681	1224
755	1156	290	12	24	99	12.182	0.2019	-1.8	1.62	1.63	2518	1247	1564	1156
755	1079	290	12	24	100	9.792	0.2071	-0.8	1.31	1.65	2381	1179	1461	1079
755	1009	290	10	28	119	13.217	0.1825	1.2	2.11	1.65	2497	1237	1356	1009
755	984	290	10	28	118	14.786	0.1773	0.3	2.36	1.64	2426	1202	1321	984
755	968	290	10	29	120	9.336	0.1770	-1.5	1.50	1.69	2485	1231	1298	968
756	1175	291	14	20	85	13.536	0.2375	1.2	1.55	1.58	2373	1175	1708	1243
756	1173	291	12	24	99	12.528	0.2019	-1.8	1.67	1.63	2553	1264	1588	1173
756	1079	291	12	24	100	10.222	0.2071	-0.8	1.37	1.64	2376	1177	1461	1079
756	1004	291	10	28	119	13.536	0.1825	1.2	2.17	1.65	2485	1231	1350	1004
756	987	291	10	29	120	9.786	0.1770	-1.5	1.57	1.69	2528	1252	1324	987
757	1187	291	14	20	85	13.847	0.2375	1.2	1.58	1.57	2397	1187	1726	1256
757	1186	291	12	24	99	12.864	0.2019	-1.8	1.72	1.62	2578	1277	1605	1186
757	1097	291	12	24	100	10.634	0.2071	-0.8	1.42	1.64	2412	1195	1486	1097
757	1002	291	10	29	120	10.215	0.1770	-1.5	1.64	1.68	2560	1268	1343	1002
757	1000	291	10	28	119	13.847	0.1775	1.2	2.22	1.65	2469	1223	1341	1000
758	1279	292	16	18	75	11.029	0.2621	-0.8	1.11	1.57	2582	1279	2013	1443
758	1204	292	14	20	85	14.150	0.2375	1.2	1.62	1.57	2430	1204	1751	1273
758	1202	292	12	24	99	13.191	0.2019	-1.8	1.77	1.62	2611	1293	1627	1202
758	1118	292	12	24	100	11.029	0.2071	-0.8	1.48	1.64	2454	1216	1515	1118
758	1018	292	10	29	120	10.627	0.1770	-1.5	1.71	1.68	2598	1287	1366	1018
758	997	292	10	28	119	14.150	0.1775	1.2	2.27	1.64	2460	1218	1337	997
758	951	292	9	32	133	11.607	0.1621	-1.0	2.08	1.69	2627	1301	1270	951
758	935	292	10	29	121	8.332	0.1821	-0.7	1.35	1.70	2424	1201	1255	935
758	928	292	9	32	132	13.191	0.1619	-1.8	2.36	1.68	2555	1265	1243	928

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	$\delta\%$	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
758	926	292	9	31	132	14.607	0.1675	1.4	2.60	1.66	2497	1237	1241	926
758	923	292	9	32	134	9.773	0.1672	-0.3	1.75	1.71	2578	1277	1235	923
759	1268	292	16	18	75	11.410	0.2621	-0.8	1.15	1.57	2561	1268	2000	1433
759	1215	292	14	20	85	14.447	0.2375	1.2	1.66	1.57	2453	1215	1768	1286
759	1212	292	12	24	99	13.509	0.2019	-1.8	1.81	1.62	2633	1304	1642	1212
759	1134	292	12	24	100	11.410	0.2071	-0.8	1.53	1.64	2485	1231	1537	1134
759	1031	292	10	29	120	11.022	0.1770	-1.5	1.78	1.68	2625	1300	1383	1031
759	992	292	10	28	119	14.447	0.1775	1.2	2.32	1.64	2449	1213	1331	992
759	955	292	10	29	121	8.832	0.1821	-0.7	1.43	1.69	2468	1223	1282	955
759	943	292	9	32	133	11.969	0.1621	-1.0	2.14	1.69	2603	1289	1260	943
759	936	292	9	32	132	13.509	0.1619	-1.8	2.41	1.67	2575	1276	1253	936
759	935	292	9	32	134	10.202	0.1672	-0.3	1.83	1.70	2606	1291	1252	935
759	933	292	9	31	132	14.894	0.1675	1.4	2.65	1.65	2514	1245	1250	933
759	870	292	9	32	135	8.061	0.1673	0.4	1.45	1.72	2457	1217	1162	870
760	1259	292	16	18	75	11.778	0.2621	-0.8	1.19	1.56	2542	1259	1989	1425
760	1225	292	14	20	85	14.737	0.2325	1.2	1.69	1.57	2472	1225	1782	1300
760	1223	292	12	24	99	13.820	0.2019	-1.8	1.85	1.61	2656	1315	1657	1223
760	1149	292	12	24	100	11.778	0.2071	-0.8	1.58	1.63	2514	1245	1557	1149
760	1042	292	10	29	120	11.402	0.1770	-1.5	1.84	1.67	2651	1313	1398	1042
760	988	292	10	28	119	14.737	0.1775	1.2	2.36	1.64	2440	1209	1326	988
760	973	292	10	29	121	9.305	0.1821	-0.7	1.50	1.69	2507	1242	1306	973
760	946	292	9	32	134	10.613	0.1672	-0.3	1.90	1.70	2632	1304	1267	946
760	943	292	9	32	132	13.820	0.1619	-1.8	2.47	1.67	2596	1286	1263	943
760	936	292	9	32	133	12.319	0.1621	-1.0	2.20	1.68	2582	1279	1251	936
760	888	292	9	32	135	8.577	0.1673	0.4	1.54	1.71	2498	1237	1186	888
761	1253	293	16	18	75	12.133	0.2621	-0.8	1.23	1.56	2529	1253	1982	1419
761	1237	293	12	24	99	14.122	0.2019	-1.8	1.90	1.61	2686	1331	1677	1237
761	1168	293	12	24	100	12.133	0.2021	-0.8	1.63	1.63	2548	1262	1579	1168
761	1056	293	10	29	120	11.770	0.1770	-1.5	1.90	1.67	2685	1330	1418	1056
761	993	293	10	29	121	9.754	0.1771	-0.7	1.58	1.69	2548	1262	1331	993
762	1249	293	12	24	99	14.418	0.1969	-1.8	1.94	1.61	2704	1339	1688	1249
762	1244	293	16	18	75	12.478	0.2571	-0.8	1.26	1.56	2511	1244	1970	1415
762	1181	293	12	24	100	12.478	0.2021	-0.8	1.68	1.63	2574	1275	1597	1181
762	1066	293	10	29	120	12.125	0.1770	-1.5	1.96	1.67	2708	1341	1432	1066
762	1008	293	10	29	121	10.182	0.1771	-0.7	1.65	1.68	2580	1278	1351	1008
763	1258	293	12	24	99	14.707	0.1969	-1.8	1.97	1.61	2725	1350	1701	1258
763	1238	293	16	18	75	12.813	0.2571	-0.8	1.29	1.56	2499	1238	1962	1409
763	1193	293	12	24	100	12.813	0.2021	-0.8	1.72	1.62	2599	1287	1614	1193
763	1072	293	10	29	120	12.470	0.1770	-1.5	2.01	1.66	2719	1347	1439	1072
763	1021	293	10	29	121	10.592	0.1771	-0.7	1.71	1.68	2610	1293	1369	1021
763	938	293	10	29	122	8.304	0.1823	0.2	1.35	1.70	2435	1206	1259	938
764	1265	294	12	24	99	14.991	0.1969	-1.8	2.02	1.60	2739	1357	1710	1265
764	1234	294	16	18	75	13.139	0.2571	-0.8	1.33	1.56	2492	1234	1958	1406
764	1209	294	12	24	100	13.139	0.2021	-0.8	1.77	1.62	2632	1304	1636	1209
764	1128	294	12	24	102	8.299	0.2125	1.2	1.13	1.66	2528	1252	1529	1128
764	1066	294	10	29	120	12.805	0.1720	-1.5	2.07	1.66	2699	1337	1429	1066
764	1037	294	10	29	121	10.986	0.1771	-0.7	1.78	1.68	2646	1310	1391	1037
764	961	294	10	29	122	8.803	0.1823	0.2	1.43	1.69	2488	1232	1290	961
765	1229	294	16	18	75	13.456	0.2571	-0.8	1.36	1.55	2482	1229	1953	1402
765	1220	294	12	24	100	13.456	0.2021	-0.8	1.81	1.62	2655	1315	1651	1220
765	1112	294	12	24	102	8.797	0.2125	1.2	1.19	1.65	2483	1230	1507	1112
765	1060	294	10	29	120	13.130	0.1720	-1.5	2.13	1.66	2681	1328	1420	1060
765	1048	294	10	29	121	11.365	0.1771	-0.7	1.84	1.68	2672	1323	1406	1048
765	979	294	10	29	122	9.274	0.1823	0.2	1.51	1.69	2527	1252	1314	979
766	1291	295	16	18	76	10.971	0.2624	0.5	1.12	1.57	2606	1291	2025	1453
766	1235	295	12	24	100	13.765	0.2021	-0.8	1.86	1.62	2686	1331	1672	1235
766	1227	295	16	18	75	13.765	0.2571	-0.8	1.40	1.55	2477	1227	1951	1400
766	1100	295	12	24	102	9.268	0.2125	1.2	1.26	1.65	2449	1213	1491	1100
766	1063	295	10	29	121	11.731	0.1771	-0.7	1.91	1.67	2705	1340	1426	1063
766	1055	295	10	29	120	13.447	0.1720	-1.5	2.18	1.66	2668	1321	1414	1055
766	999	295	10	29	122	9.722	0.1823	0.2	1.59	1.69	2571	1273	1341	999
767	1313	295	14	21	87	9.715	0.2320	-1.4	1.13	1.62	2651	1313	1817	1326
767	1280	295	16	18	76	11.350	0.2624	0.5	1.15	1.57	2584	1280	2012	1443
767	1245	295	12	24	100	14.067	0.2021	-0.8	1.90	1.61	2708	1341	1686	1245
767	1224	295	16	18	75	14.067	0.2571	-0.8	1.43	1.55	2470	1224	1947	1397

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	$\delta\%$	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
767	1089	295	12	24	102	9.715	0.2125	1.2	1.32	1.65	2418	1198	1477	1089
767	1073	295	10	29	121	12.086	0.1771	-0.7	1.97	1.67	2728	1351	1440	1073
767	1049	295	10	29	120	13.756	0.1720	-1.5	2.23	1.65	2653	1314	1407	1049
767	1014	295	10	29	122	10.148	0.1773	0.2	1.65	1.69	2600	1288	1359	1014
767	970	295	9	32	133	14.521	0.1621	-1.0	2.62	1.66	2672	1324	1299	970
767	956	295	9	32	135	11.538	0.1673	0.4	2.09	1.69	2653	1314	1279	956
767	937	295	9	32	134	13.113	0.1622	-0.3	2.37	1.68	2585	1281	1253	937
767	932	295	9	32	136	9.715	0.1675	1.2	1.76	1.71	2609	1292	1245	932
768	1298	295	14	21	87	10.142	0.2320	-1.4	1.18	1.61	2620	1298	1801	1313
768	1271	295	16	18	76	11.716	0.2624	0.5	1.19	1.57	2565	1271	2001	1434
768	1255	295	12	24	100	14.362	0.2021	-0.8	1.94	1.61	2729	1352	1700	1255
768	1219	295	16	18	75	14.362	0.2521	-0.8	1.46	1.55	2462	1219	1940	1396
768	1092	295	12	24	102	10.142	0.2125	1.2	1.38	1.64	2419	1198	1481	1092
768	1076	295	10	29	121	12.429	0.1771	-0.7	2.02	1.67	2734	1354	1444	1076
768	1044	295	10	29	120	14.058	0.1720	-1.5	2.28	1.65	2640	1308	1400	1044
768	1028	295	10	29	122	10.557	0.1773	0.2	1.72	1.68	2630	1303	1377	1028
768	978	295	9	32	133	14.806	0.1571	-1.0	2.67	1.66	2689	1332	1307	978
768	948	295	9	32	135	11.898	0.1673	0.4	2.15	1.69	2629	1302	1269	948
768	945	295	9	32	134	13.430	0.1622	-0.3	2.42	1.67	2606	1291	1264	945
768	944	295	10	29	123	8.277	0.1824	1.0	1.35	1.70	2455	1216	1266	944
768	944	295	9	32	136	10.142	0.1675	1.2	1.84	1.70	2637	1306	1262	944
768	911	295	9	33	136	8.014	0.1619	-1.9	1.45	1.72	2629	1302	1217	911
769	1286	296	14	21	87	10.550	0.2320	-1.4	1.23	1.61	2596	1286	1788	1304
769	1271	296	12	24	100	14.650	0.1971	-0.8	1.99	1.61	2755	1365	1716	1271
769	1264	296	16	18	76	12.070	0.2624	0.5	1.23	1.56	2553	1264	1994	1429
769	1237	296	16	18	75	14.650	0.2521	-0.8	1.49	1.54	2497	1237	1969	1417
769	1116	296	12	24	102	10.550	0.2075	1.2	1.44	1.64	2462	1220	1510	1116
769	1069	296	10	29	121	12.763	0.1771	-0.7	2.08	1.66	2716	1345	1436	1069
769	1043	296	10	29	122	10.950	0.1773	0.2	1.79	1.68	2666	1321	1398	1043
769	1041	296	10	29	120	14.352	0.1720	-1.5	2.34	1.65	2632	1303	1396	1041
769	968	296	10	29	123	8.774	0.1824	1.0	1.44	1.69	2508	1242	1298	968
769	958	296	9	32	136	10.550	0.1675	1.2	1.92	1.70	2671	1323	1281	958
769	955	296	9	32	134	13.738	0.1622	-0.3	2.49	1.67	2635	1305	1278	955
769	943	296	9	32	135	12.247	0.1623	0.4	2.22	1.69	2608	1292	1260	943
769	932	296	9	33	136	8.527	0.1619	-1.9	1.55	1.72	2680	1327	1245	932
770	1275	296	14	21	87	10.943	0.2320	-1.4	1.28	1.61	2573	1275	1776	1295
770	1270	296	12	24	100	14.932	0.1971	-0.8	2.02	1.60	2755	1365	1716	1270
770	1257	296	16	18	76	12.413	0.2624	0.5	1.27	1.56	2538	1257	1985	1423
770	1249	296	16	18	75	14.932	0.2521	-0.8	1.52	1.54	2522	1249	1989	1431
770	1133	296	12	24	102	10.943	0.2075	1.2	1.49	1.64	2496	1236	1533	1133
770	1064	296	10	29	121	13.087	0.1721	-0.7	2.13	1.66	2695	1335	1425	1064
770	1055	296	10	29	122	11.328	0.1773	0.2	1.85	1.68	2692	1333	1414	1055
770	1044	296	10	29	120	14.640	0.1720	-1.5	2.38	1.64	2639	1307	1400	1044
770	986	296	10	29	123	9.244	0.1824	1.0	1.51	1.69	2547	1262	1322	986
770	968	296	9	32	136	10.943	0.1675	1.2	1.99	1.70	2696	1335	1295	968
770	962	296	9	32	134	14.039	0.1622	-0.3	2.54	1.67	2655	1315	1288	962
770	948	296	9	33	136	9.010	0.1619	-1.9	1.64	1.72	2717	1346	1267	948
770	936	296	9	32	135	12.585	0.1623	0.4	2.28	1.68	2589	1282	1252	936
771	1265	297	14	21	87	11.320	0.2270	-1.4	1.33	1.61	2554	1265	1764	1290
771	1253	297	16	18	76	12.746	0.2624	0.5	1.30	1.56	2529	1253	1980	1419
771	1153	297	12	24	102	11.320	0.2075	1.2	1.55	1.64	2536	1256	1561	1153
771	1069	297	10	29	122	11.693	0.1773	0.2	1.92	1.67	2725	1350	1433	1069
771	1059	297	10	29	121	13.403	0.1721	-0.7	2.19	1.66	2682	1328	1419	1059
771	1055	297	10	29	120	14.922	0.1720	-1.5	2.43	1.64	2668	1322	1416	1055
771	1005	297	10	29	123	9.690	0.1824	1.0	1.59	1.69	2591	1283	1349	1005
771	973	297	9	32	134	14.333	0.1622	-0.3	2.60	1.66	2683	1329	1302	973
771	965	297	9	33	136	9.467	0.1619	-1.9	1.73	1.71	2759	1367	1290	965
771	964	297	9	32	136	11.320	0.1675	1.2	2.06	1.69	2680	1327	1289	964
771	939	297	9	32	135	12.914	0.1623	0.4	2.35	1.68	2594	1285	1255	939
772	1256	297	14	21	87	11.685	0.2270	-1.4	1.37	1.60	2536	1256	1756	1284
772	1245	297	16	18	76	13.070	0.2574	0.5	1.34	1.56	2514	1245	1969	1416
772	1168	297	12	24	102	11.685	0.2075	1.2	1.60	1.63	2565	1270	1581	1168
772	1079	297	10	29	122	12.046	0.1773	0.2	1.97	1.67	2749	1362	1447	1079
772	1053	297	10	29	121	13.711	0.1721	-0.7	2.24	1.65	2667	1321	1412	1053
772	1019	297	10	29	123	10.115	0.1824	1.0	1.66	1.68	2623	1299	1368	1019

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
772	979	297	9	32	134	14.621	0.1622	-0.3	2.65	1.66	2702	1338	1311	979
772	978	297	9	33	136	9.902	0.1619	-1.9	1.81	1.71	2790	1382	1307	978
772	955	297	9	32	136	11.685	0.1675	1.2	2.13	1.69	2654	1315	1279	955
772	947	297	9	32	135	13.233	0.1623	0.4	2.40	1.68	2615	1295	1266	947
773	1255	297	14	21	87	12.038	0.2270	-1.4	1.41	1.60	2534	1255	1756	1283
773	1241	297	16	18	76	13.386	0.2574	0.5	1.37	1.55	2504	1241	1963	1411
773	1181	297	12	24	102	12.038	0.2075	1.2	1.64	1.63	2592	1284	1600	1181
773	1080	297	10	29	122	12.389	0.1773	0.2	2.03	1.67	2748	1361	1449	1080
773	1048	297	10	29	121	14.012	0.1721	-0.7	2.29	1.65	2654	1315	1405	1048
773	1034	297	10	29	123	10.523	0.1774	1.0	1.73	1.68	2650	1313	1385	1034
773	990	297	9	33	136	10.318	0.1619	-1.9	1.88	1.71	2819	1396	1324	990
773	986	297	9	32	134	14.903	0.1622	-0.3	2.70	1.66	2720	1347	1320	986
773	954	297	9	32	135	13.545	0.1623	0.4	2.46	1.67	2636	1306	1276	954
773	950	297	10	29	124	8.250	0.1826	1.8	1.36	1.70	2475	1226	1274	950
773	948	297	9	32	136	12.038	0.1675	1.2	2.19	1.69	2631	1303	1268	948
774	1324	298	14	21	88	9.671	0.2322	-0.2	1.14	1.62	2673	1324	1827	1334
774	1305	298	16	18	77	10.914	0.2676	1.9	1.12	1.57	2634	1305	2040	1460
774	1275	298	14	21	87	12.381	0.2270	-1.4	1.45	1.60	2575	1275	1787	1306
774	1238	298	16	18	76	13.693	0.2574	0.5	1.40	1.55	2500	1238	1961	1409
774	1199	298	12	24	102	12.381	0.2075	1.2	1.69	1.63	2628	1302	1624	1199
774	1073	298	10	29	122	12.721	0.1773	0.2	2.09	1.66	2731	1353	1441	1073
774	1050	298	10	29	123	10.914	0.1774	1.0	1.80	1.68	2686	1331	1407	1050
774	1045	298	10	29	121	14.305	0.1721	-0.7	2.34	1.65	2646	1310	1401	1045
774	1004	298	9	33	136	10.717	0.1619	-1.9	1.96	1.70	2854	1414	1342	1004
774	974	298	10	29	124	8.746	0.1826	1.8	1.44	1.69	2528	1252	1306	974
774	965	298	9	32	135	13.849	0.1623	0.4	2.52	1.67	2665	1320	1291	965
774	942	298	9	32	136	12.381	0.1675	1.2	2.26	1.68	2614	1295	1262	942
775	1308	298	14	21	88	10.096	0.2322	-0.2	1.19	1.62	2642	1308	1810	1321
775	1294	298	16	18	77	11.291	0.2676	1.9	1.16	1.57	2612	1294	2027	1450
775	1290	298	14	21	87	12.713	0.2270	-1.4	1.49	1.60	2605	1290	1809	1322
775	1235	298	16	18	76	13.994	0.2574	0.5	1.43	1.55	2492	1235	1957	1406
775	1211	298	12	24	102	12.713	0.2075	1.2	1.74	1.63	2653	1314	1641	1211
775	1067	298	10	29	122	13.045	0.1773	0.2	2.14	1.66	2712	1343	1432	1067
775	1061	298	10	29	123	11.291	0.1774	1.0	1.86	1.68	2712	1343	1422	1061
775	1050	298	10	29	121	14.593	0.1721	-0.7	2.39	1.64	2660	1317	1409	1050
775	1005	298	9	33	136	11.101	0.1619	-1.9	2.03	1.70	2855	1414	1345	1005
775	991	298	10	29	124	9.214	0.1826	1.8	1.52	1.69	2567	1271	1330	991
775	972	298	9	32	135	14.146	0.1623	0.4	2.58	1.67	2684	1330	1301	972
775	937	298	9	32	136	12.713	0.1625	1.2	2.32	1.68	2594	1285	1252	937
776	1304	298	14	21	87	13.036	0.2270	-1.4	1.53	1.59	2633	1304	1831	1337
776	1295	298	14	21	88	10.502	0.2322	-0.2	1.24	1.61	2614	1295	1796	1311
776	1282	298	16	18	77	11.655	0.2626	1.9	1.20	1.57	2589	1282	2011	1444
776	1237	298	16	18	76	14.287	0.2574	0.5	1.46	1.55	2496	1237	1961	1409
776	1222	298	12	24	102	13.036	0.2075	1.2	1.78	1.62	2677	1326	1657	1222
776	1072	298	10	29	123	11.655	0.1774	1.0	1.92	1.67	2737	1356	1437	1072
776	1061	298	10	29	122	13.360	0.1773	0.2	2.19	1.66	2696	1335	1424	1061
776	1058	298	10	29	121	14.874	0.1721	-0.7	2.43	1.64	2679	1327	1420	1058
776	1008	298	10	29	124	9.659	0.1826	1.8	1.59	1.69	2602	1289	1352	1008
776	996	298	9	33	136	11.471	0.1619	-1.9	2.10	1.70	2825	1399	1333	996
776	979	298	9	32	135	14.436	0.1623	0.4	2.63	1.66	2703	1339	1310	979
776	946	298	9	32	136	13.036	0.1625	1.2	2.38	1.68	2616	1296	1264	946
777	1323	299	14	21	87	13.351	0.2270	-1.4	1.57	1.59	2671	1323	1859	1357
777	1285	299	14	21	88	10.893	0.2322	-0.2	1.28	1.61	2595	1285	1786	1303
777	1276	299	16	18	77	12.007	0.2626	1.9	1.24	1.56	2576	1276	2004	1439
777	1254	299	16	18	76	14.574	0.2574	0.5	1.50	1.54	2533	1254	1990	1429
777	1240	299	12	24	102	13.351	0.2025	1.2	1.83	1.62	2706	1340	1676	1240
777	1086	299	10	29	123	12.007	0.1774	1.0	1.98	1.67	2769	1372	1456	1086
777	1058	299	10	29	122	13.667	0.1723	0.2	2.25	1.65	2681	1328	1417	1058
777	1026	299	10	29	124	10.083	0.1826	1.8	1.67	1.69	2643	1309	1376	1026
777	990	299	9	33	136	11.829	0.1569	-1.9	2.17	1.69	2800	1387	1322	990
777	989	299	9	32	135	14.719	0.1623	0.4	2.69	1.66	2731	1353	1323	989
777	957	299	9	32	136	13.351	0.1625	1.2	2.44	1.68	2645	1310	1279	957
778	1334	299	14	21	87	13.658	0.2220	-1.4	1.61	1.59	2694	1334	1875	1373
778	1275	299	14	21	88	11.269	0.2322	-0.2	1.33	1.61	2575	1275	1775	1295
778	1268	299	16	18	77	12.349	0.2626	1.9	1.27	1.56	2561	1268	1996	1432

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
778	1266	299	16	18	76	14.854	0.2524	0.5	1.52	1.54	2555	1266	2008	1448
778	1250	299	12	24	102	13.658	0.2025	1.2	1.87	1.62	2728	1352	1691	1250
778	1084	299	10	29	123	12.349	0.1774	1.0	2.04	1.67	2763	1368	1454	1084
778	1052	299	10	29	122	13.966	0.1723	0.2	2.30	1.65	2668	1321	1410	1052
778	1039	299	10	29	124	10.489	0.1826	1.8	1.73	1.68	2673	1324	1394	1039
778	996	299	10	30	124	8.224	0.1769	-1.6	1.36	1.70	2652	1314	1336	996
778	995	299	9	32	135	14.997	0.1623	0.4	2.74	1.66	2749	1362	1332	995
778	983	299	9	33	136	12.175	0.1569	-1.9	2.23	1.69	2777	1375	1313	983
778	964	299	9	32	136	13.658	0.1625	1.2	2.50	1.67	2666	1320	1289	964
778	941	299	9	33	138	8.477	0.1622	-0.4	1.56	1.72	2712	1343	1256	941
779	1352	300	14	21	87	13.957	0.2220	-1.4	1.65	1.59	2729	1352	1901	1392
779	1269	300	14	21	88	11.633	0.2322	-0.2	1.38	1.60	2561	1269	1769	1290
779	1265	300	12	24	102	13.957	0.2025	1.2	1.92	1.61	2759	1367	1710	1265
779	1264	300	16	18	77	12.680	0.2626	1.9	1.31	1.56	2552	1264	1991	1429
779	1078	300	10	29	123	12.680	0.1774	1.0	2.10	1.66	2745	1360	1446	1078
779	1054	300	10	29	124	10.879	0.1826	1.8	1.80	1.68	2709	1342	1416	1054
779	1049	300	10	29	122	14.259	0.1723	0.2	2.35	1.65	2660	1317	1406	1049
779	1021	300	10	30	124	8.718	0.1770	-1.6	1.45	1.70	2708	1341	1369	1021
779	977	300	9	33	136	12.512	0.1569	-1.9	2.30	1.69	2760	1367	1306	977
779	974	300	9	32	136	13.957	0.1625	1.2	2.56	1.67	2694	1335	1303	974
779	959	300	9	33	138	8.957	0.1622	-0.4	1.65	1.72	2758	1366	1281	959
780	1364	300	14	21	87	14.250	0.2220	-1.4	1.68	1.58	2754	1364	1919	1404
780	1274	300	12	24	102	14.250	0.2025	1.2	1.96	1.61	2780	1377	1724	1274
780	1271	300	14	21	88	11.984	0.2272	-0.2	1.42	1.60	2566	1271	1773	1297
780	1258	300	16	18	77	13.003	0.2626	1.9	1.34	1.56	2541	1258	1985	1424
780	1071	300	10	29	123	13.003	0.1774	1.0	2.15	1.66	2726	1351	1437	1071
780	1067	300	10	29	124	11.255	0.1776	1.8	1.86	1.68	2732	1353	1430	1067
780	1057	300	10	29	122	14.545	0.1723	0.2	2.40	1.65	2680	1328	1417	1057
780	1040	300	10	30	124	9.184	0.1720	-1.6	1.52	1.70	2746	1360	1392	1040
780	982	300	9	33	136	12.839	0.1569	-1.9	2.36	1.69	2773	1374	1313	982
780	981	300	9	32	136	14.250	0.1625	1.2	2.61	1.67	2714	1344	1313	981
780	974	300	9	33	138	9.412	0.1622	-0.4	1.74	1.71	2792	1383	1301	974
781	1376	300	14	21	87	14.536	0.2220	-1.4	1.71	1.58	2777	1376	1936	1417
781	1335	300	14	21	89	9.628	0.2374	0.9	1.14	1.62	2695	1335	1838	1339
781	1287	300	14	21	88	12.325	0.2272	-0.2	1.46	1.60	2597	1287	1798	1315
781	1284	300	12	24	102	14.536	0.2025	1.2	2.00	1.61	2801	1387	1737	1284
781	1254	300	16	18	77	13.317	0.2626	1.9	1.37	1.55	2531	1254	1978	1419
781	1078	300	10	29	124	11.618	0.1776	1.8	1.92	1.67	2757	1366	1444	1078
781	1064	300	10	29	122	14.826	0.1723	0.2	2.44	1.64	2700	1337	1427	1064
781	1064	300	10	29	123	13.317	0.1774	1.0	2.20	1.66	2710	1342	1428	1064
781	1056	300	10	30	124	9.628	0.1719	-1.6	1.60	1.70	2783	1378	1414	1056
781	990	300	9	33	136	13.157	0.1569	-1.9	2.42	1.68	2795	1385	1324	990
781	988	300	9	32	136	14.536	0.1625	1.2	2.66	1.66	2732	1353	1322	988
781	987	300	9	33	138	9.845	0.1622	-0.4	1.81	1.71	2822	1398	1318	987
782	1392	301	14	21	87	14.816	0.2220	-1.4	1.75	1.58	2810	1392	1960	1434
782	1319	301	14	21	89	10.050	0.2324	0.9	1.19	1.62	2663	1319	1820	1329
782	1306	301	14	21	88	12.656	0.2272	-0.2	1.50	1.60	2637	1306	1827	1335
782	1281	301	12	24	102	14.816	0.2025	1.2	2.04	1.61	2795	1384	1733	1281
782	1249	301	16	18	77	13.623	0.2576	1.9	1.41	1.55	2522	1249	1973	1419
782	1201	301	12	25	104	8.203	0.2021	-1.0	1.14	1.67	2763	1369	1622	1201
782	1092	301	10	29	124	11.969	0.1776	1.8	1.99	1.67	2789	1382	1463	1092
782	1074	301	10	30	124	10.050	0.1719	-1.6	1.67	1.69	2825	1399	1439	1074
782	1060	301	10	29	123	13.623	0.1774	1.0	2.26	1.65	2698	1337	1423	1060
782	1001	301	9	33	138	10.258	0.1622	-0.4	1.90	1.71	2860	1416	1338	1001
782	1001	301	9	33	136	13.466	0.1569	-1.9	2.48	1.68	2826	1400	1339	1001
782	998	301	9	32	136	14.816	0.1625	1.2	2.72	1.66	2760	1367	1335	998
783	1320	301	14	21	88	12.978	0.2272	-0.2	1.54	1.59	2666	1320	1849	1351
783	1306	301	14	21	89	10.455	0.2324	0.9	1.24	1.61	2636	1306	1806	1319
783	1246	301	16	18	77	13.922	0.2576	1.9	1.44	1.55	2515	1246	1968	1416
783	1183	301	12	25	104	8.696	0.2021	-1.0	1.21	1.66	2714	1344	1598	1183
783	1088	301	10	30	124	10.455	0.1719	-1.6	1.74	1.69	2856	1415	1458	1088
783	1088	301	10	29	124	12.309	0.1776	1.8	2.04	1.67	2777	1375	1458	1088
783	1056	301	10	29	123	13.922	0.1724	1.0	2.31	1.65	2682	1328	1414	1056
783	1012	301	9	33	138	10.655	0.1622	-0.4	1.97	1.70	2886	1430	1353	1012
783	1009	301	9	33	136	13.769	0.1569	-1.9	2.53	1.68	2847	1410	1350	1009

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
783	1002	301	10	30	125	8.197	0.1771	-0.8	1.37	1.70	2673	1324	1343	1002
784	1339	302	14	21	88	13.291	0.2272	-0.2	1.58	1.59	2703	1339	1876	1371
784	1296	302	14	21	89	10.844	0.2324	0.9	1.29	1.61	2616	1296	1796	1311
784	1258	302	16	18	77	14.213	0.2576	1.9	1.48	1.55	2539	1258	1989	1430
784	1171	302	12	25	104	9.161	0.2021	-1.0	1.28	1.66	2676	1326	1582	1171
784	1104	302	10	30	124	10.844	0.1720	-1.6	1.81	1.69	2894	1434	1480	1104
784	1081	302	10	29	124	12.640	0.1776	1.8	2.10	1.66	2759	1367	1450	1081
784	1055	302	10	29	123	14.213	0.1724	1.0	2.36	1.65	2680	1328	1414	1055
784	1026	302	10	30	125	8.690	0.1771	-0.8	1.45	1.70	2729	1352	1376	1026
784	1020	302	9	33	136	14.064	0.1569	-1.9	2.60	1.67	2877	1425	1364	1020
784	1012	302	9	33	138	11.037	0.1622	-0.4	2.04	1.70	2883	1428	1354	1012
785	1352	302	14	21	88	13.597	0.2272	-0.2	1.61	1.59	2730	1352	1896	1385
785	1286	302	14	21	89	11.219	0.2324	0.9	1.34	1.61	2596	1286	1785	1303
785	1271	302	16	18	77	14.499	0.2576	1.9	1.50	1.54	2565	1271	2010	1445
785	1159	302	12	25	104	9.603	0.2021	-1.0	1.34	1.66	2642	1309	1566	1159
785	1116	302	10	30	124	11.219	0.1719	-1.6	1.87	1.68	2922	1447	1496	1116
785	1074	302	10	29	124	12.961	0.1776	1.8	2.16	1.66	2741	1358	1441	1074
785	1063	302	10	29	123	14.499	0.1724	1.0	2.41	1.65	2700	1338	1425	1063
785	1045	302	10	30	125	9.155	0.1771	-0.8	1.53	1.70	2770	1372	1401	1045
785	1026	302	9	33	136	14.352	0.1569	-1.9	2.65	1.67	2897	1435	1374	1026
785	1003	302	9	33	138	11.405	0.1622	-0.4	2.11	1.70	2853	1413	1342	1003
786	1365	302	14	21	88	13.895	0.2272	-0.2	1.65	1.59	2755	1365	1914	1398
786	1283	302	16	18	77	14.778	0.2576	1.9	1.53	1.54	2590	1283	2031	1460
786	1277	302	14	21	89	11.580	0.2324	0.9	1.38	1.60	2579	1277	1775	1296
786	1176	302	12	25	104	10.025	0.2021	-1.0	1.39	1.65	2676	1326	1590	1176
786	1127	302	10	30	124	11.580	0.1720	-1.6	1.93	1.68	2948	1460	1511	1127
786	1071	302	10	29	123	14.778	0.1724	1.0	2.45	1.64	2720	1347	1435	1071
786	1068	302	10	29	124	13.274	0.1776	1.8	2.21	1.66	2724	1349	1433	1068
786	1063	302	10	30	125	9.597	0.1721	-0.8	1.60	1.70	2804	1389	1422	1063
786	1033	302	9	33	136	14.634	0.1569	-1.9	2.70	1.67	2916	1445	1383	1033
786	995	302	9	33	138	11.761	0.1622	-0.4	2.18	1.69	2827	1400	1331	995
787	1380	303	14	21	88	14.186	0.2222	-0.2	1.69	1.58	2786	1380	1936	1418
787	1288	303	14	21	89	11.931	0.2324	0.9	1.42	1.60	2600	1288	1793	1308
787	1200	303	12	25	104	10.428	0.2021	-1.0	1.45	1.65	2724	1349	1622	1200
787	1141	303	10	30	124	11.931	0.1720	-1.6	1.99	1.68	2982	1477	1531	1141
787	1081	303	10	30	125	10.018	0.1721	-0.8	1.68	1.69	2847	1410	1447	1081
787	1064	303	10	29	124	13.579	0.1776	1.8	2.26	1.66	2712	1343	1427	1064
788	1392	303	14	21	88	14.471	0.2222	-0.2	1.72	1.58	2810	1392	1954	1431
788	1304	303	14	21	89	12.270	0.2324	0.9	1.46	1.60	2632	1304	1817	1325
788	1217	303	12	25	104	10.816	0.2021	-1.0	1.51	1.65	2760	1367	1647	1217
788	1206	303	12	25	105	8.171	0.2023	0.0	1.14	1.67	2780	1377	1628	1206
788	1134	303	10	30	124	12.270	0.1720	-1.6	2.05	1.67	2962	1467	1522	1134
788	1095	303	10	30	125	10.422	0.1721	-0.8	1.74	1.69	2878	1425	1465	1095
788	1059	303	10	29	124	13.877	0.1776	1.8	2.31	1.65	2699	1337	1421	1059
788	1009	303	10	30	126	8.171	0.1773	0.0	1.37	1.70	2694	1334	1351	1009
789	1403	303	14	21	88	14.750	0.2222	-0.2	1.75	1.58	2833	1403	1971	1443
789	1317	303	14	21	89	12.599	0.2274	0.9	1.50	1.60	2660	1317	1837	1344
789	1236	303	12	25	104	11.190	0.1971	-1.0	1.56	1.65	2790	1382	1667	1236
789	1189	303	12	25	105	8.662	0.2023	0.0	1.21	1.66	2730	1352	1604	1189
789	1127	303	10	30	124	12.599	0.1720	-1.6	2.10	1.67	2940	1456	1512	1127
789	1107	303	10	30	125	10.810	0.1721	-0.8	1.81	1.69	2907	1440	1483	1107
789	1057	303	10	29	124	14.168	0.1776	1.8	2.36	1.65	2694	1334	1418	1057
789	1029	303	10	30	126	8.662	0.1773	0.0	1.45	1.70	2741	1358	1380	1029
790	1336	304	14	21	89	12.920	0.2274	0.9	1.55	1.59	2698	1336	1866	1365
790	1255	304	12	25	104	11.551	0.1971	-1.0	1.61	1.64	2831	1402	1693	1255
790	1176	304	12	25	105	9.126	0.2023	0.0	1.28	1.66	2693	1334	1587	1176
790	1123	304	10	30	125	11.183	0.1721	-0.8	1.88	1.68	2943	1458	1504	1123
790	1122	304	10	30	124	12.920	0.1670	-1.6	2.16	1.67	2920	1446	1502	1122
790	1070	304	10	29	124	14.453	0.1726	1.8	2.42	1.65	2721	1348	1433	1070
790	1051	304	10	30	126	9.126	0.1773	0.0	1.53	1.70	2791	1382	1409	1051
791	1350	304	14	21	89	13.232	0.2274	0.9	1.58	1.59	2726	1350	1887	1380
791	1269	304	12	25	104	11.900	0.1971	-1.0	1.66	1.64	2860	1417	1713	1269
791	1164	304	12	25	105	9.566	0.2023	0.0	1.34	1.66	2658	1317	1572	1164
791	1134	304	10	30	125	11.544	0.1721	-0.8	1.94	1.68	2969	1471	1519	1134
791	1116	304	10	30	124	13.232	0.1670	-1.6	2.21	1.66	2902	1437	1494	1116

C	P_{min}	F	m_n	N_1	N_2	ψ	$x_1/-x_2$	δ%	m_F	m_p	P_{ac1}	P_{ac2}	P_{at1}	P_{at2}
791	1077	304	10	29	124	14.731	0.1726	1.8	2.46	1.64	2741	1358	1443	1077
791	1068	304	10	30	126	9.566	0.1773	0.0	1.61	1.70	2828	1401	1432	1068
792	1368	305	14	21	89	13.536	0.2274	0.9	1.62	1.59	2762	1368	1913	1399
792	1287	305	12	25	104	12.239	0.1971	-1.0	1.72	1.64	2897	1435	1738	1287
792	1188	305	12	25	105	9.987	0.2023	0.0	1.40	1.66	2708	1341	1605	1188
792	1148	305	10	30	125	11.893	0.1721	-0.8	2.00	1.68	3003	1487	1538	1148
792	1111	305	10	30	124	13.536	0.1670	-1.6	2.27	1.66	2889	1431	1488	1111
792	1086	305	10	30	126	9.987	0.1773	0.0	1.68	1.69	2871	1422	1457	1086
793	1381	305	14	21	89	13.833	0.2274	0.9	1.66	1.59	2788	1381	1932	1412
793	1299	305	12	25	104	12.567	0.1971	-1.0	1.76	1.64	2924	1448	1755	1299
793	1207	305	12	25	105	10.389	0.2023	0.0	1.46	1.65	2746	1360	1631	1207
793	1139	305	10	30	125	12.231	0.1721	-0.8	2.06	1.67	2977	1475	1527	1139
793	1106	305	10	30	124	13.833	0.1670	-1.6	2.32	1.66	2875	1424	1481	1106
793	1101	305	10	30	126	10.389	0.1723	0.0	1.75	1.69	2899	1436	1474	1101
794	1393	305	14	21	89	14.123	0.2274	0.9	1.69	1.58	2812	1393	1950	1425
794	1312	305	12	25	104	12.887	0.1971	-1.0	1.80	1.63	2950	1461	1772	1312
794	1225	305	12	25	105	10.775	0.2023	0.0	1.51	1.65	2782	1378	1655	1225
794	1210	305	12	25	106	8.140	0.2074	1.0	1.15	1.67	2801	1387	1637	1210
794	1131	305	10	30	125	12.560	0.1721	-0.8	2.11	1.67	2955	1464	1517	1131
794	1114	305	10	30	126	10.775	0.1723	0.0	1.82	1.69	2928	1450	1491	1114
794	1108	305	10	30	124	14.123	0.1670	-1.6	2.37	1.66	2881	1427	1485	1108
795	1410	306	14	21	89	14.407	0.2274	0.9	1.73	1.58	2846	1410	1974	1442
795	1327	306	12	25	104	13.198	0.1971	-1.0	1.85	1.63	2984	1478	1794	1327
795	1246	306	12	25	105	11.148	0.2023	0.0	1.57	1.65	2825	1399	1684	1246
795	1194	306	12	25	106	8.630	0.2074	1.0	1.22	1.66	2754	1364	1615	1194
795	1130	306	10	30	126	11.148	0.1723	0.0	1.88	1.68	2965	1469	1512	1130
795	1125	306	10	30	125	12.879	0.1721	-0.8	2.17	1.67	2938	1455	1509	1125
795	1120	306	10	30	124	14.407	0.1670	-1.6	2.42	1.65	2912	1442	1501	1120
796	1420	306	14	21	89	14.684	0.2224	0.9	1.76	1.58	2866	1420	1987	1456
796	1339	306	12	25	104	13.502	0.1971	-1.0	1.90	1.63	3008	1490	1810	1339
796	1260	306	12	25	105	11.507	0.2023	0.0	1.62	1.64	2856	1415	1705	1260
796	1181	306	12	25	106	9.092	0.2024	1.0	1.28	1.66	2709	1342	1594	1181
796	1140	306	10	30	126	11.507	0.1723	0.0	1.94	1.68	2991	1481	1527	1140
796	1128	306	10	30	124	14.684	0.1670	-1.6	2.47	1.65	2933	1453	1512	1128
796	1119	306	10	30	125	13.190	0.1721	-0.8	2.22	1.66	2920	1446	1501	1119
797	1436	307	14	21	89	14.956	0.2224	0.9	1.80	1.58	2899	1436	2010	1473
797	1355	307	12	25	104	13.798	0.1921	-1.0	1.94	1.62	3037	1504	1827	1355
797	1281	307	12	25	105	11.855	0.1973	0.0	1.67	1.64	2892	1432	1728	1281
797	1175	307	12	25	106	9.530	0.2024	1.0	1.35	1.66	2688	1331	1586	1175
797	1152	307	10	30	126	11.855	0.1723	0.0	2.01	1.68	3018	1495	1543	1152
797	1139	307	10	30	124	14.956	0.1670	-1.6	2.52	1.65	2963	1468	1527	1139
797	1115	307	10	30	125	13.493	0.1671	-0.8	2.28	1.66	2904	1439	1493	1115
798	1463	307	16	19	79	10.748	0.2521	-1.0	1.14	1.59	2954	1463	2207	1591
798	1366	307	12	25	104	14.088	0.1921	-1.0	1.98	1.62	3060	1516	1842	1366
798	1295	307	12	25	105	12.192	0.1973	0.0	1.72	1.64	2920	1446	1747	1295
798	1195	307	12	25	106	9.949	0.2024	1.0	1.41	1.66	2730	1352	1614	1195
798	1143	307	10	30	126	12.192	0.1723	0.0	2.06	1.67	2993	1482	1531	1143
798	1110	307	10	30	125	13.790	0.1671	-0.8	2.33	1.66	2890	1431	1486	1110
798	1021	307	10	30	128	8.120	0.1776	1.6	1.38	1.71	2736	1355	1367	1021
799	1451	307	16	19	79	11.120	0.2521	-1.0	1.18	1.58	2929	1451	2193	1580
799	1367	307	12	25	104	14.371	0.1921	-1.0	2.02	1.62	3062	1517	1844	1367
799	1308	307	12	25	105	12.520	0.1973	0.0	1.77	1.64	2947	1460	1765	1308
799	1215	307	12	25	106	10.350	0.2024	1.0	1.46	1.65	2769	1371	1641	1215
799	1135	307	10	30	126	12.520	0.1723	0.0	2.12	1.67	2970	1471	1521	1135
799	1115	307	10	30	125	14.079	0.1671	-0.8	2.38	1.66	2903	1438	1493	1115
799	1042	307	10	30	128	8.608	0.1776	1.6	1.46	1.70	2782	1378	1395	1042
800	1440	308	16	19	79	11.478	0.2471	-1.0	1.22	1.58	2907	1440	2178	1575
800	1362	308	12	25	104	14.647	0.1921	-1.0	2.07	1.62	3052	1512	1838	1362
800	1324	308	12	25	105	12.839	0.1973	0.0	1.82	1.63	2982	1477	1787	1324
800	1237	308	12	25	106	10.735	0.2024	1.0	1.52	1.65	2814	1394	1671	1237
800	1129	308	10	30	126	12.839	0.1723	0.0	2.18	1.67	2953	1463	1514	1129
800	1127	308	10	30	125	14.362	0.1671	-0.8	2.43	1.65	2934	1453	1509	1127
800	1064	308	10	30	128	9.069	0.1776	1.6	1.55	1.70	2833	1403	1425	1064

Appendix C Subroutine Names and Their Purposes

NOTE: UNDERSCORE IN MODULE NAMES SHOULD BE IGNORED FOR ALPHABETICAL ORDER.

MODULE NAME	BRIEF DESCRIPTION OF PURPOSE OF THE MODULE
ACTIVITY	MAIN CONTROL PROGRAM FOR GEAR/SHAFT + DESIGN/CHECK
AINV	ARC-INVOLUTE FUNCTION
ANG_CRS	CENTRE VECTOR DIRECTIONAL ANGLE WHEN COORDINATES & CENTRES KNOWN
APPG_INPUT	APPLICATION DATA INPUT FOR GEARS
APPG_INPUT_CHK	CHECK THAT APPLICATION DATA ARE IN VALID RANGES(GEAR)
APPS_INPUT	APPLICATION DATA INPUT FOR SHAFTS
APPS_INPUT_CHK	CHECK THAT APPLICATION DATA ARE IN VALID RANGES(SHAFT)
BACKLASH	BACKLASHES OF GEARS
BANKIN	DEPOSIT A DESIGN IN DESIGN BANK
BANKOUT	WITHDRAW A DESIGN FROM DESIGN BANK
BEAMR_PRE	PRE-TREATMENT FOR BEAMR_FEM
BEAMR_FEM	MULTI-SUPPORT BEAM REACTION BY FINITE ELEMENT METHOD
BLOCK DATA	INTERPOLATING DATA FOR EBGD
BND_KT	STRESS CONCENTRATION FACTOR FOR BENDING
BRG2_DSN	TWO BEARING SHAFT DESIGN
BRG2_INIT	TWO BEARING SHAFT INITIALISATION
BRG3_DSN	THREE BEARING SHAFT DESIGN
BRG3_INIT	THREE BEARING SHAFT INITIALISATION
BRG_AXL_CHK	BEARING AXIAL DIMENSION CONSISTENCY CHECK
BRG_BORE_CHK	BEARING BORE AND GEAR ROOT DIAMETER COMPATIBILITY CHECK
BRG_FIND	FIND BEARING DESIGNATION BY BEARING DIMENSIONS
BRG_LIFE	BEARING LIFE CALCULATION
BRG_OD_CHK	BEARING O/D INTERFERENCE CHECK
BRGS_DJ	DETERMINATION OF SHAFT SEGMENT DIAMETER BY BEARING
BRGS_JNL	BEARING JOURNAL DIMENSIONS
CALC_D1I	DETERMINE PINION P.C.D. BY SURFACE DURABILITY
CALC_D1J	DETERMINE PINION P.C.D. BY BENDING STRENGTH
CALC_RATIOS	CALCULATE DEFAULT GEAR RATIOS
C_H2	HARDNESS RATIO FACTOR
CHECK_SHAFT	MAIN PROGRAM FOR SHAFT CHECK CALCULATIONS
CHK_BRGS	TEST IF BEARING DIMENSIONS AND LIVES OF TWO ITERATIONS ARE SAME
CLEAR	CLEAR SCREEN
CL_KL	LIFE FACTORS
CM_KM	LOAD DISTRIBUTION FACTORS
COM_INIT	COMPACT(COMBINED) DRIVE DRIVING OUTPUT SHAFT INITIALISATION
CREAMING	KEEP ONLY THE BEST TWO DESIGNS FOR EACH MODULE
CR_KR	RELIABILITY FACTORS
CRS_FACE	FACE WIDTH WHEN CENTRE DISTANCE AND FACE TO P.C.D RATIO KNOWN
CV_KV	DYNAMIC FACTORS
DATA_MAN	DATA MANAGEMENT(READ, SAVE AND CLOSE DATA FILES)
DEGREE	CONVERT RADIAN TO DEGREE
DEL_JNLS	DELETE A SEGMENT/JOURNAL
DESIGN_GEAR	MAIN PROGRAM FOR GEAR DESIGN CALCULATION
DESIGN_SHAFT	MAIN PROGRAM FOR SHAFT DESIGN CALCULATION
DRIVES	MASTER PROGRAM OF THE 'DRIVES' PACKAGE
DRVG_CHK	MILL PINION DRIVING SHAFT CHECK
DRVG_DSN	MILL PINION DRIVING SHAFT DESIGN
DRVG_INIT	MILL PINION DRIVING SHAFT INITIALISATION
DRVN_CHK	MILL PINION DRIVEN SHAFT CHECK
DRVN_DSN	MILL PINION DRIVEN SHAFT DESIGN
DRVN_INIT	MILL PINION DRIVEN SHAFT INITIALISATION
EBGD	FABRICATED WHEEL DIMENSIONS(PER BS 436-1940)
EXT_AXL_CHK	EXTERNAL LOAD AXIAL POSITION CONSISTENCY CHECK
FIX_BRG	DETERMINE WHICH BEARING AS FIXED
GAPS	GRINDING ALLOWANCE PER FLANK
GEAR_AXL_CHK	GEAR AXIAL DIMENSION CONSISTENCY CHECK
GEARC_RPT	MAIN PROGRAM FOR GEAR RATING RESULT REPORT
GEARC_RPTF	REPORT FULL DETAIL RESULTS FOR GEAR RATING
GEARC_RPTM	REPORT MAIN RESULTS FOR GEAR RATING
GEAR_DESIGN	MAIN CONTROL PROGRAM FOR GEAR DESIGN
GEARDIM	GEAR DIMENSION CALCULATION
GEARD_RPT	MAIN PROGRAM FOR GEAR DESIGN RESULT REPORT
GEARD_RPTC	REPORT CONSTRAINTS USED IN DESIGN
GEARD_RPTF	REPORT FULL DETAIL RESULTS FOR GEAR DESIGN
GEARD_RPTM	REPORT MAIN RESULTS FOR GEAR DESIGN
GEARD_RPTO	REPORT ALL OPTIONAL DESIGNS FOR GEAR DESIGN

GEAR_EXTRA	EXTRA GEAR DATA INPUT AFTER COPYING EXISTING GEAR DATA
GEAR_INPUT	GEAR DATA INPUT
GEAR_INPUT_CHK	CHECK THAT GEAR DATA ARE IN VALID RANGES
GEARLIM	CHECK IF ANY BOUNDARY IS VIOLATED
GEAR_RATING	MAIN CONTROL PROGRAM FOR GEAR RATING
GEOM	GEOMETRY FACTORS FOR A DESIGN
GEOM_EGTH	GEOMETRY FACTORS I AND J PER AGMA 323.01
GET_BDR_VAL	INPUT BOUNDARY VALUES
GET_BRGS	INPUT BEARING DATA
GET_BRGS_CHK	CHECK THAT BEARING DATA ARE IN VALID RANGES
GET_CENTRES	INPUT CENTRE DISTANCES
GET_CENTRES_CHK	CHECK THAT CENTRE DISTANCES ARE IN VALID RANGE
GET_EXTL	INPUT EXTERNAL LOADS
GET_EXTL_CHK	CHECK THAT EXTERNAL LOADS ARE IN VALID RANGE
GET_EXTQ	INPUT EXTERNAL TORQUES
GET_EXTQ_CHK	CHECK THAT EXTERNAL TORQUES ARE IN VALID RANGE
GET_FACES	INPUT FACE WIDTHS
GET_FACES_CHK	CHECK THAT FACE WIDTHS ARE IN VALID RANGE
GET_FACTORS	INPUT RATING FACTORS
GET_GMATER	INPUT GEAR MATERIALS
GET_GMATER_CHK	CHECK THAT GEAR MATERIAL VALUES ARE IN VALID RANGES
GET_GRPAR	INPUT GEAR PARAMETERS
GET_GRPAR_CHK	CHECK THAT GEAR PARAMETERS ARE IN VALID RANGES
GET_GRPOS	INPUT GEAR AXIAL POSITION
GET_GRPOS_CHK	CHECK THAT GEAR AXIAL POSITIONS ARE IN VALID RANGE
GET_GRTYPE	INPUT GEAR TYPES(SPUR, HELICAL, DOUBLE HELICAL)
GET_GRTYPE_CHK	CHECK THAT GEAR TYPES ARE OF VALID TYPES
GET_HELIX	INPUT HAND OF HELIX
GET_JNLS	INPUT SHAFT SEGMENT/JOURNAL DATA
GET_JNLS_CHK	CHECK THAT SHAFT SEGMENT/JOURNAL DATA ARE IN VALID RANGES
GET_KEYS	INPUT KEY DATA
GET_KEYS_CHK	CHECK THAT KEY DATA ARE IN VALID RANGES
GET_MACHN	INPUT MACHINING VALUES
GET_MOD	INPUT MODULES TO USE
GET_OPT_VAL	OTHER OPTIONAL VALUE INPUT
GET_PRATIOS	INPUT PINION PROPORTIONS
GET_PRECI	INPUT PRECISION VALUES
GET_PROPS	INPUT GEAR TOOTH PROPORTIONS
GET_PW	INPUT POWER(KW OR HP)
GET_RATIOS	INPUT REQUIRED GEAR RATIOS
GET_RATIOS_CHK	CHECK THAT RATIOS ARE IN VALID RANGES
GET_SF	INPUT SERVICE FACTORS
GET_SKMATER	INPUT SHAFT AND KEY MATERIALS
GET_STD	CHOOSE STANDARD TO USE(AGMA 323.01, 6005-B89, 2001-B88)
GET_STD_CHK	CHECK THAT A STANDARD HAS BEEN CHOSEN
GET_STD_VAL	INPUT STANDARD VALUES
GET_STRS	INPUT STRESSED SECTION DATA
GET_STRS_CHK	CHECK THAT STRESSED SECTION DATA ARE IN VALID RANGE
GET_VECTOR	INPUT CENTRE VECTOR
GET_VECTOR_CHK	CHECK THAT CENTRE VECTORS ARE IN VALID RANGE
GET_X1X2	INPUT ADDENDUM MODIFICATION COEFFICIENTS
GMATER2001	GEAR MATERIAL VALUE FOR AGMA 2001-B88
GMATER323	GEAR MATERIAL VALUE FOR AGMA 323.01
GMATER6005	GEAR MATERIAL VALUE FOR AGMA 6005-B89
GMIJ	GEOMETRY FACTORS I AND J PER AGMA 908-B89
HEADING	HEADING ON REPORT FILE
HLPMSG	DISPLAY HELP MESSAGE
HUNTING	NUMBER OF HUNTING TEETH
I_KMATER	INDEX OF KEY MATERIAL
INIT_DATA	DATA INITIALISATION BY SYSTEM/TEMPORARY/USER DATA FILES
INIT_SHF	MAIN PROGRAM FOR SHAFT INITIALISATION
INP_CHK	INPUT SHAFT CHECK
INP_DSN	INPUT SHAFT DESIGN
INP_FILE	USER DATA FILE NAME FOR INPUT DATA(INITIALISE DATA)
INP_INIT	INPUT SHAFT INITIALISATION
INPUT_A	INPUT ANGULAR VARIABLES
INPUT_I	INPUT INTER VARIABLES
INPUT_O	INPUT OPTION LIST VARIABLES
INPUT_R	INPUT REAL VARIABLES
INPUT_TAB	SELECT AN OPTION FROM A LIST
INPUT_YN	INPUT YES/NO VARIABLES
INS_JNLS	INSERT A SEGMENT/JOURNAL
INT_CHK	INTERMEDIATE SHAFT CHECK
INT_DSN	INTERMEDIATE SHAFT DESIGN
INT_INIT	INTERMEDIATE SHAFT INITIALISATION
INT_JNL	INTER SEGMENT BETWEEN PINION AND WHEEL ON INTERMEDIATE SHAFT

I_SMATER	INDEX OF SHAFT MATERIAL
ITOSTR	INTEGER CONVERTED TO STRING
JNL_AXL_CHK	SHAFT SEGMENT/JOURNAL AXIAL DIMENSION CONSISTENCY CHECK
JNL_RD	SHAFT RADIUS BETWEEN SEGMENTS
JNLR_POS	PRINT SHAFT SEGMENT/JOURNAL RADIUS AT ITS AXIAL POSITION
K_B2	RIM THICKNESS FACTOR
KEY_AXL_CHK	KEY AXIAL DIMENSION CONSISTENCY CHECK
KEY_BHL	INITIALISE KEY DIMENSIONS BY SHAFT DIAMETER
KEYD_CHK	CHECK: KEYED DIAMETER LARGER THAN ALL THE 'THROUGH' DIAMETERS
KEYS_DJ	DETERMINATION OF SHAFT SEGMENT DIAMETER BY KEY
KT_ALL	STRESS CONCENTRATION FACTORS FOR BENDING, TORSION AND TENSION
LASER	LASER PRINTER DRIVER
LOAD_SHR	LOAD SHARING CONDITION TEST FOR SPUR GEARS, PER AGMA 908-B89
LOCAL	LOCAL PRINTER DRIVER
LOC_POS	LOCATE POSITION INDEX ON A MULTI-SEGMENT SHAFT
MAX_D1I	MAXIMUM PINION P.C.D. DETERMINED BY SURFACE DURABILITY
MAX_D1J	MAXIMUM PINION P.C.D. DETERMINED BY BENDING STRENGTH
MAX_I	MAXIMUM GEOMETRY FACTOR I FOR SURFACE DURABILITY
MIN_CRS	MICRO MINIMISED CENTRE DISTANCE
MIN_D1I	MINIMUM PINION P.C.D. DETERMINED BY SURFACE DURABILITY
MIN_D1J	MINIMUM PINION P.C.D. DETERMINED BY BENDING STRENGTH
MIN_I	MINIMUM GEOMETRY FACTOR I FOR SURFACE DURABILITY
MIN_M	MINIMUM MODULE DETERMINED BY BENDING STRENGTH
MSG_FILE	MESSAGE FILE NAME
NLEN	ACTUAL LENGTH OF A CHARACTER STRING
OPT_INPUT	OPTIONAL DATA INPUT
OUT_ALLOW	OUTPUT MATERIAL ALLOWABLES
OUT_BRGC	OUTPUT BEARING RATINGS
OUT_BRGD	OUTPUT BEARING DIMENSIONS
OUT_BRGL	OUTPUT BEARING LOADS
OUT_CHK	OUTPUT SHAFT CHECK
OUT_DFLXN	OUTPUT DEFLECTIONS OVER GEAR FACES
OUT_DSN	OUTPUT SHAFT DESIGN
OUT_EXTL	OUTPUT EXTERNAL LOADS
OUT_EXTQ	OUTPUT EXTERNAL TORQUES
OUT_FACTOR	OUTPUT GEAR RATING FACTORS
OUT_FILE	USER DATA FILE NAME FOR OUTPUT DATA(SAVE DATA)
OUT_FIXC	OUTPUT MAIN RESULTS FOR FIXED CENTRE DISTANCE
OUT_GEARD1	OUTPUT MAIN RESULTS OF ONE DESIGN
OUT_GEARD9	OUTPUT MAIN PARAMETER LIST OF MANY DESIGNS
OUT_GEARP	OUTPUT GEAR PARAMETERS
OUT_GEARSZ	OUTPUT GEAR SIZES
OUT_GPPS	OUTPUT GEAR PITCH POINT DATA
OUT_INIT	OUTPUT SHAFT INITIALISATION
OUT_JNLS	OUTPUT SHAFTS SEGMENT/JOURNAL DATA
OUT_KEYD	OUTPUT KEY DIMENSIONS
OUT_KT	OUTPUT STRESS CONCENTRATION FACTORS
OUT_LOWC	OUTPUT MAIN RESULTS FOR INADEQUATE CENTRE DISTANCE
OUT_MANFC	OUTPUT GEAR MANUFACTURING DATA
OUT_MATRL	OUTPUT GEAR MATERIALS
OUT_MINC	OUTPUT MAIN RESULTS FOR MINIMISED CENTRE DISTANCE
OUT_RATING	OUTPUT GEAR POWER RATINGS
OUT_STRESS	OUTPUT GEAR TOOTH STRESSES
OUT_TRNSM	OUTPUT TRANSMISSION CONDITIONS
OUT_USTR	OUTPUT UTILISATION RATIOS AND STRESSES
OUT_WHLD	OUTPUT GEAR WHEEL DIMENSIONS
OUT_WHLW	OUTPUT GEAR WHEEL WEIGHTS AND GD2'S
OUT_XYZG	OUTPUT GEAR LOAD COMPONENTS
PART_POS	PRINT PART NAME ON SHAFT AT ITS AXIAL POSITION
PAUSE	PAUSE FOR USER TO HIT [RETURN]
PPP	PINION PITCH POINT COORDINATES
PRINTER	PRINTER CONTROL PROGRAM
PROTUB	AMOUNT OF PROTUBERANCE OF GEARS
PRT_FILE	PRINT FILE NAME
RADIAN	CONVERT DEGREE TO RADIAN
RATE_GEAR	MAIN PROGRAM FOR GEAR RATING CALCULATION
RATING	POWER RATINGS FOR A DESIGN
RD_KEY	RETRIEVE DATA FROM KEY AND KEYWAY TABLE
RDL_AXL	RADIAL AND AXIAL LOAD FOR BEARING
REDN_INPUT	REDUCTION DATA INPUT
REDN_INPUT_CHK	CHECK THAT REDUCTION DATA ARE IN VALID RANGES
REPT_MESG	REPORT WARNING AND DIAGNOSTIC MESSAGES
RG1CRS00	GEAR DESIGN FOR FIXED RATIO AND FREE CENTRE DISTANCE
RG1CRS11	GEAR DESIGN FOR FIXED RATIO AND FIXED CENTRE DISTANCE
RPM_HH_VCT	INITILISATION OF RPM DIRECTION, HAND OF HELIX, CENTRE VECTOR
RPT_FILE	REPORT FILE NAME

```
R_TO_R          RUN TIME DESIGN 'B' COPIED TO RUN TIME DESIGN 'A'
R_TO_T          RUN TIME DESIGN 'A' COPIED TO TEMPORARY DESIGN
SAC2001         AGMA 2001-B88 ALLOWABLES FOR SURFACE DURABILITY
SAC6005         AGMA 6005-B89 ALLOWABLES FOR SURFACE DURABILITY
SAT2001         AGMA 2001-B88 ALLOWABLES FOR BENDING STRENGTH
SAT6005         AGMA 6005-B89 ALLOWABLES FOR BENDING STRENGTH
SAVE_BRGS       STORE BEARING DIMENSIONS AND LIVES
SAVE_DATA       SAVE DATA FILE
SAVE_TEMP       SAVE TEMPORARY DATA FILE
SAVE_USER       SAVE USER DATA FILE
SAY2001         AGMA 2001-B88 ALLOWABLES FOR YIELD STRENGTH
SEC_AXL_CHK     STRESSED SECTION AXIAL POSITION CONSISTENCY CHECK
SEC_MNT         MOMENT AT A SECTION
SEC_TQ          TORQUE AT A SECTION
SELECT          LIST ALL DESIGNS AND ASK USER TO SELECT ONE
SHAFT_CHECK     MAIN CONTROL PROGRAM FOR SHAFT CHECK
SHAFT_DESIGN    MAIN CONTROL PROGRAM FOR SHAFT DESIGN
SHAFT_WT        SHAFT WEIGHT AND GD2
SHF_AXL_CHK     MAIN PROGRAM FOR SHAFT AXIAL DIMENSION CONSISTENCY CHECK
SHFC_OUT        REPORT INDIVIDUAL SHAFT CHECK RESULTS
SHFC_RPT        MAIN PROGRAM FOR SHAFT CHECK RESULT REPORT
SHFD_OUT        REPORT INDIVIDUAL SHAFT DESIGN RESULTS
SHFD_RPT        MAIN PROGRAM FOR SHAFT DESIGN RESULT REPORT
SHF_INIT        INDIVIDUAL SHAFT INITIALISATION
SHF_INPUT       MAIN PROGRAM FOR SHAFT DATA INPUT
SHF_INPUT_CHK   CHECK THAT SHAFT DATA ARE IN VALID RANGES
SHF_KODE        SHAFT CALCULATION HISTORY CODE SUMMARY
SHF_LIST        LIST OF SHAFT NAMES
SHF_LOAD        SHAFT REACTIONS AND DEFLECTIONS
SHF_SPEC        SHAFT DATA INPUT
SKF_SRB1        AUTO-SELECTION OF SKF SPHERICAL ROLLER BEARING
SKF_SRB5        SELECTION OF SKF SPHERICAL ROLLER BEARINGS FOR 5 SERIES
SKF_SRB9        MANUAL-SELECTION OF SKF SPHERICAL ROLLER BEARING
SKF_SRBD        LIST BEARINGS SELECTED BY SKF_SRB5 AND ASK USER TO PICK ONE
SKF_SRBS        SELECTION OF SKF SPHERICAL ROLLER BEARING BY SERIES
SORTING         ARRANGE DESIGN INDEX IN THE ORDER OF THE HIGHEST POWER THE FIRST
SRB_FIND        FIND BEARING DESIGNATION FOR SKF SPHERICAL ROLLER BEARING
SRB_LIFE        SKF SPHERICAL ROLLER BEARING LIFE
STEPLEFT        ONE INCREMENT STEP TO THE LEFT OF THE CURRENT CENTRE DISTANCE
STRS_DJ         DETERMINATION OF SHAFT SEGMENT DIAMETER BY STRENGTH
TDI_FIND(DUMMY) FIND BEARING DESIGNATION FOR TIMKEN TAPER ROLLER BEARING TDI
TDI_LIFE(DUMMY) TIMKEN TAPER ROLLER TDI BEARING LIFE
TDO_FIND(DUMMY) FIND BEARING DESIGNATION FOR TIMKEN TAPER ROLLER BEARING TDO
TDO_LIFE(DUMMY) TIMKEN TAPER ROLLER TDO BEARING LIFE
TEM_FILE        TEMPORARY DATA FILE NAME
TIMKEN_TDI(DUMMY)      SELECTION OF TIMKEN TAPER ROLLER BEARING TDI
TIMKEN_TDO(DUMMY)      SELECTION OF TIMKEN TAPER ROLLER BEARING TDO
TOLEFT          THE LEFT MOST POINT OF THE CENTRE DISTANCE RANGE OF A DESIGN
TORIGHT         THE RIGHT MOST POINT OF THE CENTRE DISTANCE RANGE OF A DESIGN
TQ_AXL_CHK      EXTERNAL TORQUE AXIAL POSITION CONSISTENCY CHECK
TRIM            TRIM PRECEDING AND TRAILING SPACES OFF A STRING
TSN_KT          STRESS CONCENTRATION FACTOR FOR TORSION
T_TO_R          TEMPORARY DESIGN COPIED TO RUN TIME DESIGN 'B'
UNDERTOP        UNDERCUT AND TOP LAND CONDITION CHECK
UNDO_BITS       MARK BITS IN CALCULATION HISTORY CODE AS 'UNDO'
UTL_STRS        STRESSES AND UTILISATION RATIOS
VPITCHM         ALLOWABLE PITCH ERROR PER AGMA 2000-A88
VPITSHR         MAXIMUM PITCH ERROR FOR LOAD SHARING PER AGMA 908-B89
WHEEL           FABRICATED WHEEL DIMENSIONS
WHL_OD_CHK      WHEEL O/D INTERFERENCE CHECK
WMI             WEIGHT AND GD2 OF FABRICATED WHEEL
WPP             WHEEL PITCH POINT COORDINATES
X1X2_ADD4       ALLOCATION OF ADDENDUM MODIFICATION COEFFICIENTS BY MAAG METHOD
X1X2_HA         HELIX ANGLE WHEN ADDENDUM MODIFICATION COEFFICIENTS KNOWN
X1X2_Y          CENTRE DISTANCE MODIF. COEF. WHEN ADDENDUM MODIF. COEF. KNOWN
ZPART           AXIAL POSITION OF PART ON SHAFT
```

Appendix D Full Expansion of Program Calling Structure

The conventions used are:

Parallel structure:

```
|
|----MODULE_NAME_1
|
|----MODULE_NAME_2
|
```

meaning one of the modules is selected to be called;

Sequential structure:

```
|
|
-----MODULE_NAME
|
|
```

meaning the module is always called;

Expandable Module:

MODULE_NAME(...)

meaning other modules are called by the named module.

The Root and Trunk

```
DRIVES                             master program of the package
|
-----HLPMSG                        display help message
|
-----INIT_DATA(...)                initialise data
|
|----DATA_MAN                      data management
|     |
|     |----INIT_DATA(...)          initialise data
|     |
|     |----SAVE_USER(...)          save user data file
|
|----ACTIVITY                      desired action
      |
      |----SAVE_USER(...)          save user data file
      |
      |----GEAR_DESIGN(...)        main module: gear design
      |
      |----SHAFT_DESIGN(...)       main module: shaft design
      |
      |----GEAR_RATING(...)        main module: gear rating
      |
      |----SHAFT_CHECK(...)        main module: shaft check
```

Branch GEAR_DESIGN

```
GEAR_DESIGN
|
-----GET_STD_CHK                standard chosen check
|
-----APPG_INPUT_CHK             application data check(gear)
|
-----REDN_INPUT_CHK(...)        reduction data check
|
|----SAVE_USER(...)             save user data file
|
|----GET_STD                    choose a gear standard
|
|----APPG_INPUT(...)            application data input(gear)
|
|----REDN_INPUT(...)            reduction data input
|
|----OPT_INPUT(...)             optional data input
|
|----DESIGN_GEAR(...)           gear design calculation
|
|----GEARD_RPT(...)             gear design result report

REDN_INPUT_CHK                  check that reduction data are in valid ranges
|
-----GET_RATIOS_CHK             check that ratios are in valid range
|
-----GET_GRTYPE_CHK             check that gear types are of valid types
|
-----GET_CENTRES_CHK            check that centre distances are in valid range
|
-----GET_FACES_CHK              check that face widths are in valid range
|
-----GET_GMATER_CHK             check that material values are in valid range

SAVE_USER                       save user data file
|
-----OUT_FILE                   user data file name for output data(save data)
|
-----SAVE_DATA                  save data file

APPG_INPUT                      application data input for gears
|
-----GET_PW                     input power(kw or hp)
|
-----GET_SF                     input service factors
```

Branch GEAR_DESIGN(continued)

```
REDN_INPUT                     reduction data input
|
-----GET_RATIOS_CHK            check that ratios are in valid range
|
|----GET_RATIOS                input required gear ratios
|    |
|    -----CALC_RATIOS          calculate default gear ratios
|
|----GET_GRTYPE                input gear types(spur, helical, double helical)
|
|----GET_CENTRES               input centre distances
|
|----GET_FACES                 input face widths
|
|----GET_X1X2                  input addendum modification coefficients
|
|----GET_GMATER                input gear materials
     |
     |----GMATER323            gear material value for agma 323.01
     |
     |----GMATER6005           gear material value for agma 6005-b89
     |    |
     |    -----SAC6005         agma 6005-b89 allowables for surface durability
     |    |
     |    -----SAT6005         agma 6005-b89 allowables for bending strength
     |    |
     |    -----SAY2001         agma 2001-b88 allowables for bending yield
     |
     |----GMATER2001           gear material value for agma 2001-b88
          |
          -----SAC2001         agma 2001-b88 allowables for surface durability
          |
          -----SAT2001         agma 2001-b88 allowables for bending strength
          |
          -----SAY2001         agma 2001-b88 allowables for bending yield

OPT_INPUT                      optional data input
|
|----GET_BDR_VAL               input boundary values
|
|----GET_STD_VAL               input standard values
|    |
|    |----GET_MOD              input modules to use
|    |
|    |----GET_PROPS            input gear tooth proportions
|
|----GET_OPT_VAL               other optional value input
     |
     |----GET_MACHN            input machining values
     |
     |----GET_PRECI            input precision values
     |
     |----GET_PRATIOS          input pinion proportions
     |
     |----GET_FACTORS          input rating factors
```

Branch GEAR_DESIGN(continued)

```
DESIGN_GEAR                   main program for gear design calculation
|
-----SAVE_TEMP                save temporary data file
|    |
|    -----SAVE_DATA           save data file
|
-----CR_KR                    reliability factors
|
-----RG1CRS00                 gear design for fixed ratio and free centre distance
     |
     -----C_H2                hardness ratio factor
     |
     -----CL_KL               life factors
     |
     -----MIN_D1I             minimum pinion p.c.d. determined by surface durability
     |    |
     |    -----MAX_I          maximum geometry factor i for surface durability
     |    |
     |    -----CALC_D1I       determine pinion p.c.d. by surface durability
     |         |
     |         -----CV_KV     dynamic factors
     |         |
     |         -----CM_KM     load distribution factors
     |
     -----MAX_D1I             maximum pinion p.c.d. determined by surface durability
     |    |
     |    -----MIN_I          minimum geometry factor i for surface durability
     |    |
     |    -----CALC_D1I       determine pinion p.c.d. by surface durability
     |         |
     |         -----CV_KV     dynamic factors
     |         |
     |         -----CM_KM     load distribution factors
     |
     -----MIN_D1J             minimum pinion p.c.d. determined by bending strength
     |    |
     |    -----CALC_D1J       determine pinion p.c.d. by bending strength
     |         |
     |         -----CV_KV     dynamic factors
     |         |
     |         -----CM_KM     load distribution factors
     |
     -----MAX_D1J             maximum pinion p.c.d. determined by bending strength
     |    |
     |    -----CALC_D1J       determine pinion p.c.d. by bending strength
     |         |
     |         -----CV_KV     dynamic factors
     |         |
     |         -----CM_KM     load distribution factors
     |
     -----CRS_FACE            face width when centre distance & aspect ratio known
     |
     -----RG1CRS11(...)       gear design for fixed ratio and fixed centre distance
     |
     -----MIN_CRS(...)        micro minimised centre distance
     |
     -----OUT_MINC            output main results for minimised centre distance
     |
     -----OUT_GEARD1          output main results of one design
```

Branch GEAR_DESIGN(continued)

```
RG1CRS11                       gear design for fixed ratio and fixed centre distance
|
-----MIN_M                     minimum module determined by bending strength
|    |
|    -----CM_KM                load distribution factors
|    |
|    -----CV_KV                dynamic factors
|
-----X1X2_Y            centre distance modif. coef. when addendum modif. coef. known
|
-----GEARDIM                   gear dimension calculation
|    |
|    -----X1X2_ADD4            addendum modification coefficients by maag method
|    |
|    -----X1X2_HA              helix angle when addendum modification coef'ts known
|
-----RATING(...)               power ratings for a design
|
-----CREAMING                  keep only the best two designs for each module
|    |
|    -----R_TO_T               run time design 'a' copied to temporary design
|    |
|    -----R_TO_R               run time design 'b' copied to run time design 'a'
|    |
|    -----T_TO_R               temporary design copied to run time design 'b'
|
-----BANKIN                    deposit a design in design bank
|
|-----
|    |
|    -----SORTING      index designs in the order of the highest power the first
|    |
|    -----SELECT       list all designs and ask user to select one
|    |    |
|    |    -----OUT_GEARD9      output main parameter list of many designs
|    |    |    |
|    |    |    -----HUNTING    number of hunting teeth
|    |    |
|    |    ------BANKOUT        withdraw a design from design bank
|    |
|    -----OUT_FIXC             output main results for fixed centre distance
|         |
|         -----OUT_GEARD1      output main results of one design
|              |
|              -----HUNTING    number of hunting teeth
|-----
|    |
|    -----SORTING      index designs in the order of the highest power the first
|    |
|    -----BANKOUT      withdraw a design from design bank
|    |
|    -----OUT_LOWC     output main results for inadequate centre distance
|         |
|         -----OUT_GEARD1      output main results of one design
|              |
|              -----HUNTING    number of hunting teeth
|-----
     |
     -----SORTING      index designs in the order of the highest power the first
     |
     -----BANKOUT      withdraw a design from design bank
```

Branch GEAR_DESIGN(continued)

```
RATING                            power ratings for a design
|
-----GEOM                         geometry factors for a design
|     |
|     |----GEOM_EGTH              geometry factors i and j per agma 323.01
|     |
|     |----|
|          -----BACKLASH          backlashes
|          |
|          -----GAPS              grinding allowance per flank
|          |
|          -----PROTUB            amount of protuberance
|          |
|          -----LOAD_SHR          load sharing test for spur gears(agma 908-b89)
|          |     |
|          |     -----VPITCHM     allowable pitch error per agma 2000-a88
|          |     |
|          |     -----VPITSHR     maximum pitch error for load sharing(908-b89)
|          |
|          -----GMIJ              geometry factors i and j per agma 908-b89
|                |
|                -----AINV        arc-involute function
|
-----CV_KV                        dynamic factors
|
-----CM_KM                        load distribution factors


MIN_CRS                           micro minimised centre distance
|
-----CRS_FACE                     face width when centre distance & aspect ratio known
|
-----|
|    -----RG1CRS11(...)           gear design for fixed ratio and fixed centre distance
|    |
|    -----BANKOUT                 withdraw a design from design bank
|    |
|    -----TOLEFT       the left most point of the centre distance range of a design
|    |     |
|    |     -----CRS_FACE          face width when centre distance and aspect ratio known
|    |
|    -----GEARDIM                 gear dimension calculation
|    |     |
|    |     -----X1X2_ADD4         addendum modification coefficients by maag method
|    |     |
|    |     -----X1X2_HA           helix angle when addendum modification coef'ts known
|    |
|    -----RATING(...)             power ratings for a design
|
-----|
     -----RG1CRS11(...)           gear design for fixed ratio and fixed centre distance
     |
     -----BANKOUT                 withdraw a design from design bank
     |
     -----TORIGHT      the right most point of the centre distance range of a design
     |     |
     |     -----CRS_FACE          face width when centre distance and aspect ratio known
     |
     -----GEARDIM                 gear dimension calculation
     |     |
     |     -----X1X2_ADD4         addendum modification coefficients by maag method
     |     |
     |     -----X1X2_HA           helix angle when addendum modification coef'ts known
     |
     -----RATING(...)             power ratings for a design
```

Branch GEAR_DESIGN(continued)

```
GEARD_RPT                        main program for gear design result report
|
-----RPT_FILE                    report file name
|
-----HEADING                     heading on report file
|
|----GEARD_RPTM                  report main results for gear design
|    |
|    -----OUT_GEARD1             output main results of one design
|         |
|         -----HUNTING           number of hunting teeth
|
|----GEARD_RPTF                  report full detail results for gear design
|    |
|    -----OUT_TRNSM              output transmission conditions
|    |
|    -----OUT_GEARP              output gear parameters
|    |    |
|    |    -----HUNTING           number of hunting teeth
|    |
|    -----OUT_GEARSZ             output gear sizes
|    |
|    -----OUT_MATRL              output gear materials
|    |
|    -----OUT_MANFC              output gear manufacturing data
|    |
|    -----OUT_RATING             output gear power ratings
|    |
|    -----OUT_STRESS             output gear tooth stresses
|    |
|    -----OUT_FACTOR             output gear rating factors
|
|----GEARD_RPTO                  report all optional designs for gear design
|    |
|    -----OUT_GEARD9             output main parameter list of many designs
|         |
|         -----HUNTING           number of hunting teeth
|
|----GEARD_RPTC                  report constraints used in design
|
|----REPT_MESG                   report warning and diagnostic messages
|    |
|    -----MSG_FILE               message file name
|
|----PRT_FILE                    report file name
|
-----PRINTER                     printer control program
     |
     |----LASER                  laser printer driver
     |
     |----LOCAL                  local printer driver
          |
          -----NLEN              actual length of a character string
```

Branch SHAFT_DESIGN

```
SHAFT_DESIGN
|
-----RPM_HH_VCT              init:direction of rotation,hand of helix,centre vector
|
-----APPS_INPUT_CHK          application data check(shaft)
|
-----GEAR_INPUT_CHK(...)     gear data check
|
-----INIT_SHF                initialisation of shafts
|    |
|    -----SHF_INIT(...)      initialisation of each shaft
|
-----SHF_INPUT_CHK(...)      shaft data check
|
-----SHF_AXL_CHK(...)        shaft axial position check
|
-----SHF_KODE                shaft calculation history
|
|----SAVE_USER(...)          save user data file
|
|----APPS_INPUT(...)         application data input(shaft)
|
|----GEAR_INPUT(...)         existing gear data input
|
|----SHF_INPUT(...)          shaft data input
|
|----DESIGN_SHAFT(...)       shaft design calculation
|
|----SHFD_RPT(...)           shaft design result report


GEAR_INPUT_CHK               check that gear data are in valid ranges
|
-----GET_GRTYPE_CHK          check that gear types are of valid types
|
-----GET_CENTRES_CHK         check that centre distances are in valid range
|
-----GET_FACES_CHK           check that face widths are in valid range
|
-----GET_GRPAR_CHK           check that gear parameters are in valid ranges
|
------
     |
     |----GET_GMATER_CHK     check that gear material values are in valid ranges
     |
     |-----
          |
          -----GET_VECTOR_CHK check that centre vectors are in valid ranges
          |
          -----GET_GRPOS_CHK  check that gear axial positions are in valid range

SHF_INIT                     individual shaft initialisation
|
|----INP_INIT                input shaft initialisation
|
|----INT_INIT                intermediate shaft initialisation
|    |
|    |----BRG3_INIT          three bearing shaft initialisation
|    |
|    |----BRG2_INIT          two bearing shaft initialisation
|
|----COM_INIT         compact(combined) drive driving output shaft initialisation
|    |
|    |----BRG3_INIT          three bearing shaft initialisation
|    |
|    |----BRG2_INIT          two bearing shaft initialisation
|
|----OUT_INIT                output shaft initialisation
|
|----DRVG_INIT               mill pinion driving shaft initialisation
|
|----DRVN_INIT               mill pinion driven shaft initialisation
```

Branch SHAFT_DESIGN(continued)

```
SHF_INPUT_CHK                  check that shaft data are in valid ranges
|
-----GET_EXTL_CHK              check that external loads are in valid range
|
-----GET_EXTQ_CHK              check that external torques are in valid range
|
-----GET_JNLS_CHK              check that shaft segment data are in valid ranges
|
-----GET_STRS_CHK              check that stressed section data are in valid ranges
|
-----GET_BRGS_CHK              check that bearing data are in valid ranges
|
-----GET_KEYS_CHK              check that key data are in valid ranges

SHF_AXL_CHK            main program for shaft axial dimension consistency check
|
-----JNL_AXL_CHK               shaft segment axial dimension consistency check
|
-----GEAR_AXL_CHK              gear axial dimension consistency check
|
-----BRG_AXL_CHK               bearing axial dimension consistency check
|
-----EXT_AXL_CHK               external load axial position consistency check
|
-----TQ_AXL_CHK                external torque axial position consistency check
|
-----KEY_AXL_CHK               key axial dimension consistency check
|
-----SEC_AXL_CHK               stressed section axial position consistency check

SAVE_USER                      save user data file
|
-----OUT_FILE                  user data file name for output data(save data)
|
-----SAVE_DATA                 save data file

APPS_INPUT                     application data input for shafts
|
-----GET_PW                    input power(kw or hp)

GEAR_INPUT                     gear data input
|
|----GEAR_EXTRA                extra gear data input after copying existing gear data
|    |
|    -----RPM_HH_VCT           init of rpm direction, hand of helix, centre vector
|    |
|    |----GET_VECTOR           input centre vector
|    |
|    |----GET_GRPOS            input gear axial position
|    |
|    |----GET_HELIX            input hand of helix
|
|----|
     -----RPM_HH_VCT           init of rpm direction, hand of helix, centre vector
     |
     -----GET_CENTRES_CHK      check that centre distances are in valid range
     |
     -----GET_FACES_CHK        check that face widths are in valid range
     |
     |----GET_GRTYPE           input gear types(spur, helical, double helical)
     |
     |----GET_GRPAR            input gear parameters
     |
     |----GET_CENTRES          input centre distances
     |
     |----GET_FACES            input face widths
     |
     |    |----GET_X1X2        input addendum modification coefficients
     |----|
     |    |----GET_VECTOR      input centre vector
     |
     |    |----GET_GMATER      input gear materials
     |----|
          |----GET_GRPOS       input gear axial position
```

Branch SHAFT_DESIGN(continued)

```
SHF_INPUT                      main program for shaft data input
|
-----SHF_LIST                  list of shaft names
|
-----SHF_SPEC                  shaft data input
     |
     -----SHF_INIT(...)        individual shaft initialisation
     |
     |----GET_BRGS             input bearing data
     |    |
     |    -----BRG_FIND        find bearing designation by bearing dimensions
     |         |
     |         |----SRB_FIND   for skf spherical roller bearing
     |         |
     |         |----TDI_FIND   (DUMMY) for timken taper roller bearing tdi
     |         |
     |         |----TDO_FIND(DUMMY)   for timken taper roller bearing tdo
     |
     |----GET_EXTL             input external loads
     |
     |----GET_EXTQ             input external torques
     |
     |----GET_JNLS             input shaft segment/journal data
     |    |
     |    -----SHF_INIT(...)   individual shaft initialisation
     |    |
     |    -----INS_JNLS        insert a segment/journal
     |    |
     |    -----DEL_JNLS        delete a segment/journal
     |    |
     |    -----GET_SKMATER     input shaft and key materials
     |         |
     |         |----I_SMATER   index of shaft material
     |         |
     |         |----I_KMATER   index of key material
     |         |
     |         -----INPUT_TAB  select an option from a list
     |
     |----GET_STRS             input stressed section data
     |    |
     |    -----SHF_INIT(...)   individual shaft initialisation
     |
     |----GET_KEYS             input key data
          |
          -----SHF_INIT(...)   individual shaft initialisation
          |
          -----KEY_BHL         initialise key dimensions by shaft diameter
          |
          -----GET_SKMATER     input shaft and key materials
               |
               |----I_SMATER   index of shaft material
               |
               |----I_KMATER   index of key material
               |
               -----INPUT_TAB  select an option from a list
```

Branch SHAFT_DESIGN(continued)

```
DESIGN_SHAFT                  main program for shaft design calculation
|
-----SAVE_TEMP                save temporary data file
|    |
|    -----SAVE_DATA           save data file
|
-----SHF_LIST                 list of shaft names
|
|----INP_DSN(...)             input shaft design
|
|----INT_DSN                  intermediate shaft design
|    |
|    |----BRG2_DSN(...)       two bearing shaft design
|    |
|    |----BRG3_DSN(...)       three bearing shaft design
|
|----OUT_DSN(...)             output shaft design
|
|----DRVG_DSN(...)            mill pinion driving shaft design
|
|----DRVN_DSN(...)            mill pinion driven shaft design
|
-----BRG_BORE_CHK             bearing bore and gear root diameter compatibility check
|
-----BRG_OD_CHK               bearing o/d interference check
|
-----WHL_OD_CHK               wheel o/d interference check


SHF_LOAD                      shaft reactions and deflections
|
-----BEAMR_PRE                pre-treatment for beamr_fem
|
-----BEAMR_FEM                multi-support beam reaction by finite element method
|
-----LOC_POS                  locate position index on a multi-segment shaft

STRS_DJ                       determination of shaft segment diameter by strength
|
-----BND_KT                   stress concentration factor for bending
|    |
|    -----LOC_POS             locate position index on a multi-segment shaft
|    |
|    -----KT_ALL              stress factors for bending, torsion and tension
|
-----TSN_KT                   stress concentration factor for torsion
|    |
|    -----LOC_POS             locate position index on a multi-segment shaft
|    |
|    -----KT_ALL              stress factors for bending, torsion and tension
|
-----SEC_TQ                   torque at a section
|
-----SEC_MNT                  moment at a section
|
-----LOC_POS                  locate position index on a multi-segment shaft

KEYS_DJ                       determination of shaft segment diameter by key
|
-----SEC_TQ                   torque at a section
|
-----LOC_POS                  locate position index on a multi-segment shaft
|
-----RD_KEY                   retrieve data from key and keyway table
```

Branch SHAFT_DESIGN(continued)

```
BRGS_DJ                         determination of shaft segment diameter by bearing
|
-----FIX_BRG                    determine which bearing as fixed
|
-----LOC_POS                    locate position index on a multi-segment shaft
|
-----RDL_AXL                    radial and axial load for bearing
|
|-----
|    |
|    |----SKF_SRB1              auto-selection of skf spherical roller bearing
|    |    |
|    |    -----SKF_SRB5         selection of skf spherical roller bearings for 5 series
|    |         |
|    |         -----SKF_SRBS      selection of skf spherical roller bearing by series
|    |
|    |----SKF_SRB9              manual-selection of skf spherical roller bearing
|         |
|         -----SKF_SRB5         selection of skf spherical roller bearings for 5 series
|         |    |
|         |    -----SKF_SRBS    selection of skf spherical roller bearing by series
|         |
|         -----SKF_SRBD         list bearings selected by skf_srb5,ask user to pick one
|
|----TIMKEN_TDI(DUMMY)          selection of timken taper roller bearing tdi
|
|----TIMKEN_TDO(DUMMY)          selection of timken taper roller bearing tdo

BRGS_JNL              bearing journal dimensions
|
-----LOC_POS          locate position index on a multi-segment shaft

KEYD_CHK              check: keyed diameter larger than all the 'through' diameters
|
-----LOC_POS          locate position index on a multi-segment shaft

INT_JNL               inter segment between pinion and wheel on intermediate shaft
|
-----LOC_POS          locate position index on a multi-segment shaft

JNL_RD                shaft radius between segments
|
-----LOC_POS          locate position index on a multi-segment shaft

WHEEL                 fabricated wheel dimensions
|
-----LOC_POS          locate position index on a multi-segment shaft
|
-----EBGD             fabricated wheel dimensions(per bs 436-1940)
     |
     -----WMI         weight and gd2 of fabricated wheel

UTL_STRS              stresses and utilisation ratios
|
-----BND_KT           stress concentration factor for bending
|    |
|    -----LOC_POS     locate position index on a multi-segment shaft
|    |
|    -----KT_ALL      stress factors for bending, torsion and tension
|
-----TSN_KT           stress concentration factor for torsion
|    |
|    -----LOC_POS     locate position index on a multi-segment shaft
|    |
|    -----KT_ALL      stress factors for bending, torsion and tension
|
-----LOC_POS          locate position index on a multi-segment shaft
|
-----SEC_TQ           torque at a section
|
-----SEC_MNT          torque at a section
```

Branch SHAFT_DESIGN(continued)

```
INP_DSN                        input shaft design
|
-----PPP                       pinion pitch point coordinates
|
-----SHAFT_WT                  shaft weight and gd2
|
-----SHF_LOAD(...)             shaft reactions and deflections
|
-----STRS_DJ(...)              determination of shaft segment diameter by strength
|
-----KEYS_DJ(...)              determination of shaft segment diameter by key
|
-----BRGS_DJ(...)              determination of shaft segment diameter by bearing
|
-----BRGS_JNL(...)             bearing journal dimensions
|
-----KEYD_CHK(...)      check: keyed diameter larger than all the 'through' diameters
|
-----JNL_RD(...)               shaft radius between segments
|
-----UTL_STRS(...)             stresses and utilisation ratios
|
-----SAVE_TEMP                 save temporary data file
     |
     -----SAVE_DATA            save data file

BRG2_DSN                       two bearing shaft design
|
-----PPP                       pinion pitch point coordinates
|
-----WPP                       wheel pitch point coordinates
|
-----SHAFT_WT                  shaft weight and gd2
|
-----SHF_LOAD(...)             shaft reactions and deflections
|
-----STRS_DJ(...)              determination of shaft segment diameter by strength
|
-----KEYS_DJ(...)              determination of shaft segment diameter by key
|
-----BRGS_DJ(...)              determination of shaft segment diameter by bearing
|
-----BRGS_JNL(...)             bearing journal dimensions
|
-----INT_JNL(...)       inter segment between pinion and wheel on intermediate shaft
|
-----KEYD_CHK(...)      check: keyed diameter larger than all the 'through' diameters
|
-----WHEEL(...)                fabricated wheel dimensions
|
-----JNL_RD(...)               shaft radius between segments
|
-----UTL_STRS(...)             stresses and utilisation ratios
|
-----SAVE_TEMP                 save temporary data file
     |
     -----SAVE_DATA            save data file
```

Branch SHAFT_DESIGN(continued)

```
BRG3_DSN                        three bearing shaft design
|
-----PPP                        pinion pitch point coordinates
|
-----WPP                        wheel pitch point coordinates
|
-----SHAFT_WT                   shaft weight and gd2
|
-----SHF_LOAD(...)              shaft reactions and deflections
|
-----STRS_DJ(...)               determination of shaft segment diameter by strength
|
-----KEYS_DJ(...)               determination of shaft segment diameter by key
|
-----SAVE_BRGS                  store bearing dimensions and lives
|
-----BRGS_DJ(...)               determination of shaft segment diameter by bearing
|
-----BRGS_JNL(...)              bearing journal dimensions
|
-----CHK_BRGS           test if bearing dimensions and lives of two iterations are same
|
-----WHEEL(...)                 fabricated wheel dimensions
|
-----KEYD_CHK(...)      check: keyed diameter larger than all the 'through' diameters
|
-----JNL_RD(...)                shaft radius between segments
|
-----UTL_STRS(...)              stresses and utilisation ratios
|
-----SAVE_TEMP                  save temporary data file
     |
     -----SAVE_DATA             save user data file

OUT_DSN                         output shaft design
|
-----WPP                        wheel pitch point coordinates
|
-----SHAFT_WT                   shaft weight and gd2
|
-----SHF_LOAD(...)              shaft reactions and deflections
|
-----STRS_DJ(...)               determination of shaft segment diameter by strength
|
-----KEYS_DJ(...)               determination of shaft segment diameter by key
|
-----BRGS_DJ(...)               determination of shaft segment diameter by bearing
|
-----BRGS_JNL(...)              bearing journal dimensions
|
-----KEYD_CHK(...)  check: keyed diameter larger than all the 'through' diameters
|
-----WHEEL(...)                 fabricated wheel dimensions
|
-----JNL_RD(...)                shaft radius between segments
|
-----UTL_STRS(...)              stresses and utilisation ratios
|
-----SAVE_TEMP                  save temporary data file
     |
     -----SAVE_DATA             save data file
```

Branch SHAFT_DESIGN(continued)

```
DRVG_DSN                        mill pinion driving shaft design
|
-----PPP                        pinion pitch point coordinates
|
-----SHAFT_WT                   shaft weight and gd2
|
-----SHF_LOAD(...)              shaft reactions and deflections
|
-----STRS_DJ(...)               determination of shaft segment diameter by strength
|
-----KEYS_DJ(...)               determination of shaft segment diameter by key
|
-----BRGS_DJ(...)               determination of shaft segment diameter by bearing
|
-----BRGS_JNL(...)              bearing journal dimensions
|
-----KEYD_CHK(...)      check: keyed diameter larger than all the 'through' diameters
|
-----JNL_RD(...)                shaft radius between segments
|
-----UTL_STRS(...)              stresses and utilisation ratios
|
-----SAVE_TEMP                  save temporary data file
     |
     -----SAVE_DATA             save data file

DRVN_DSN                        mill pinion driven shaft design
|
-----WPP                        wheel pitch point coordinates
|
-----SHAFT_WT                   shaft weight and gd2
|
-----SHF_LOAD(...)              shaft reactions and deflections
|
-----STRS_DJ(...)               determination of shaft segment diameter by strength
|
-----KEYS_DJ(...)               determination of shaft segment diameter by key
|
-----BRGS_DJ(...)               determination of shaft segment diameter by bearing
|
-----BRGS_JNL(...)              bearing journal dimensions
|
-----KEYD_CHK(...)      check: keyed diameter larger than all the 'through' diameters
|
-----JNL_RD(...)                shaft radius between segments
|
-----UTL_STRS(...)              stresses and utilisation ratios
|
-----SAVE_TEMP                  save temporary data file
     |
     -----SAVE_DATA             save data file
```

Branch SHAFT_DESIGN(continued)

```
SHFD_RPT                      main program for shaft design result report
|
-----SHF_LIST                 list of shaft names
|
-----RPT_FILE                 report file name
|
-----HEADING                  heading on report file
|
|----SHFD_OUT                 report individual shaft design results
|    |
|    -----SHF_LIST            list of shaft names
|    |
|    -----OUT_EXTQ            output external torques
|    |
|    -----OUT_JNLS            output shafts segment data
|    |    |
|    |    -----ZPART          axial position of part on shaft
|    |    |
|    |    -----PART_POS       print part name on shaft at its axial position
|    |    |
|    |    -----JNLR_POS       print shaft segment radius at its axial position
|    |
|    -----OUT_KEYD            output key dimensions
|    |
|    -----OUT_GPPS            output gear pitch point data
|    |
|    -----OUT_XYZG            output gear load components
|    |
|    -----OUT_EXTL            output external loads
|    |
|    -----OUT_BRGL            output bearing loads
|    |
|    -----OUT_BRGD            output bearing dimensions
|    |
|    -----OUT_BRGC            output bearing ratings
|    |
|    -----OUT_USTR            output utilisation ratios and stresses
|    |
|    -----OUT_KT              output stress concentration factors
|    |
|    -----OUT_DFLXN           output deflections over gear faces
|    |
|    -----OUT_ALLOW           output material allowables
|    |
|    -----OUT_WHLD            output gear wheel dimensions
|    |
|    -----OUT_WHLW            output gear wheel weights and gd2's
|
|----REPT_MESG                report warning and diagnostic messages
|
|----PRT_FILE                 print file name
|    |
|    --------MSG_FILE         message file name
|
-----PRINTER                  printer control program
     |
     |----LASER               laser printer driver
     |
     |----LOCAL               local printer driver
          |
          -----NLEN           actual length of a character string
```

Branch GEAR_RATING

```
GEAR_RATING
|
-----GET_STD_CHK              standard chosen check
|
-----APPG_INPUT_CHK           application data check(gear)
|
-----GEAR_INPUT_CHK(...)      gear data check
|
|----SAVE_USER(...)           save user data file
|
|----GET_STD                  choose a gear standard
|
|----APPG_INPUT(...)          application data input(gear)
|
|----GEAR_INPUT(...)          existing gear data input
|
|----OPT_INPUT(...)           optional data input
|
|----RATE_GEAR(...)           gear rating calculation
|
|----GEARC_RPT(...)           gear rating result report


GEAR_INPUT_CHK                check that gear data are in valid ranges
|
-----GET_GRTYPE_CHK           check that gear types are of valid types
|
-----GET_CENTRES_CHK          check that centre distances are in valid range
|
-----GET_FACES_CHK            check that face widths are in valid range
|
-----GET_GRPAR_CHK            check that gear parameters are in valid ranges
|
------
     |
     |----GET_GMATER_CHK      check that gear material values are in valid ranges
     |
     |-----
          |
          -----GET_VECTOR_CHK check that centre vectors are in valid ranges
          |
          -----GET_GRPOS_CHK   check that gear axial positions are in valid ranges

SAVE_USER                     save user data file
|
-----OUT_FILE                 user data file name for output data(save data)
|
-----SAVE_DATA                save data file

APPG_INPUT                    application data input for gears
|
-----GET_PW                   input power(kw or hp)
|
-----GET_SF                   input service factors
```

Branch GEAR_RATING(continued)

```
GEAR_INPUT                     gear data input
|
|----GEAR_EXTRA                extra gear data input after copying existing gear data
|    |
|    -----RPM_HH_VCT           init of rpm direction, hand of helix, centre vector
|    |
|    |----GET_VECTOR           input centre vector
|    |
|    |----GET_GRPOS            input gear axial position
|    |
|    |----GET_HELIX            input hand of helix
|
|-----
     |
     -----RPM_HH_VCT           init of rpm direction, hand of helix, centre vector
     |
     -----GET_CENTRES_CHK      check that centre distances are in valid range
     |
     -----GET_FACES_CHK        check that face widths are in valid range
     |
     |----GET_GRTYPE           input gear types(spur, helical, double helical)
     |
     |----GET_GRPAR            input gear parameters
     |
     |----GET_CENTRES          input centre distances
     |
     |----GET_FACES            input face widths
     |
     |-----
     |    |
     |    |----GET_X1X2        input addendum modification coefficients
     |    |
     |    |----GET_VECTOR      input centre vector
     |
     |-----
          |
          |----GET_GMATER      input gear materials
          |
          |----GET_GRPOS       input gear axial position

OPT_INPUT                      optional data input
|
|----GET_BDR_VAL               input boundary values
|
|----GET_STD_VAL               input standard values
|    |
|    |----GET_MOD              input modules to use
|    |
|    |----GET_PROPS            input gear tooth proportions
|
|----GET_OPT_VAL               other optional value input
     |
     |----GET_MACHN            input machining values
     |
     |----GET_PRECI            input precision values
     |
     |----GET_PRATIOS          input pinion proportions
     |
     |----GET_FACTORS          input rating factors
```

Branch GEAR_RATING(continued)

```
RATE_GEAR                     main program for gear rating calculation
|
-----SAVE_TEMP                save temporary data file
|    |
|    -----SAVE_DATA           save data file
|
-----CR_KR                    reliability factors
|
-----C_H2                     hardness ratio factor
|
-----GEARDIM                  gear dimension calculation
|    |
|    -----X1X2_ADD4           addendum modification coefficients by maag method
|    |
|    -----X1X2_HA             helix angle when àddendum modification coef'ts known
|
-----RATING(...)              power ratings for a design
|
-----GEARLIM                  check if any boundary is violated
|    |
|    -----UNDERTOP            undercut and top land condition check
|
-----GEARC_RPTM               report main results for gear rating
     |
     -----OUT_GEARD1          output main results of one design
          |
          -----HUNTING        number of hunting teeth

RATING                        power ratings for a design
|
-----GEOM                     geometry factors for a design
|    |
|    |----GEOM_EGTH           geometry factors i and j per agma 323.01
|    |
|    |----|
|         -----BACKLASH       backlashes
|         |
|         -----GAPS           grinding allowance per flank
|         |
|         -----PROTUB         amount of protuberance
|         |
|         -----LOAD_SHR       load sharing test for spur gears(agma 908-b89)
|         |    |
|         |    -----VPITCHM   allowable pitch error per agma 2000-a88
|         |    |
|         |    -----VPITSHR   maximum pitch error for load sharing(908-b89)
|         |
|         -----GMIJ           geometry factors i and j per agma 908-b89
|              |
|              -----AINV      arc-involute function
|
-----CV_KV                    dynamic factors
|
-----CM_KM                    load distribution factors
```

Branch GEAR_RATING(continued)

```
GEARC_RPT                     main program for gear rating result report
|
-----RPT_FILE                 report file name
|
-----HEADING                  heading on report file
|
|----GEARC_RPTM               report main results for gear rating
|    |
|    -----OUT_GEARD1          output main results of one design
|         |
|         -----HUNTING        number of hunting teeth
|
|----GEARC_RPTF               report full detail results for gear rating
|    |
|    -----OUT_TRNSM           output transmisión conditions
|    |
|    -----OUT_GEARP           output gear parameters
|    |    |
|    |    -----HUNTING        number of hunting teeth
|    |
|    -----OUT_GEARSZ          output gear sizes
|    |
|    -----OUT_MATRL           output gear materials
|    |
|    -----OUT_MANFC           output gear manufacturing data
|    |
|    -----OUT_RATING          output gear power ratings
|    |
|    -----OUT_STRESS          output gear tooth stresses
|    |
|    -----OUT_FACTOR          output gear rating factors
|
|----REPT_MESG                report warning and diagnostic messages
|    |
|    -----MSG_FILE            message file name
|
|----PRT_FILE                 report file name
|
-----PRINTER                  printer control program
     |
     |----LASER               laser printer driver
     |
     |----LOCAL               local printer driver
          |
          -----NLEN           actual length of a character string
```

Branch SHAFT_CHECK

```
SHAFT_CHECK
|
-----RPM_HH_VCT             init:direction of rotation,hand of helix,centre vector
|
-----APPS_INPUT_CHK         application data check(shaft)
|
-----GEAR_INPUT_CHK(...)    gear data check
|
-----INIT_SHF               initialisation of shafts
|    |
|    -----SHFT_INIT(...)    initialisation of each shaft
|
-----SHF_INPUT_CHK(...)     shaft data check
|
-----SHF_KODE               shaft calculation history
|
|----SAVE_USER(...)         save user data file
|
|----APPS_INPUT(...)        application data input(shaft)
|
|----GEAR_INPUT(...)        existing gear data input
|
|----SHF_INPUT(...)         shaft data input
|
|----CHECK_SHAFT(...)       shaft check calculation
|
|----SHFC_RPT(...)          shaft check result report


GEAR_INPUT_CHK              check that gear data are in valid ranges
|
-----GET_GRTYPE_CHK         check that gear types are of valid types
|
-----GET_CENTRES_CHK        check that centre distances are in valid range
|
-----GET_FACES_CHK          check that face widths are in valid range
|
-----GET_GRPAR_CHK          check that gear parameters are in valid ranges
|
------
     |
     |----GET_GMATER_CHK    check that gear material values are in valid ranges
     |
     |-----
          |
          -----GET_VECTOR_CHK check that centre vectors are in valid range
          |
          -----GET_GRPOS_CHK   check that gear axial positions are in valid range

SHF_INIT                    individual shaft initialisation
|
|----INP_INIT               input shaft initialisation
|
|----INT_INIT               intermediate shaft initialisation
|    |
|    |----BRG3_INIT         three bearing shaft initialisation
|    |
|    |----BRG2_INIT         two bearing shaft initialisation
|
|----COM_INIT       compact(combined) drive driving output shaft initialisation
|    |
|    |----BRG3_INIT         three bearing shaft initialisation
|    |
|    |----BRG2_INIT         two bearing shaft initialisation
|
|----OUT_INIT               output shaft initialisation
|
|----DRVG_INIT              mill pinion driving shaft initialisation
|
|----DRVN_INIT              mill pinion driven shaft initialisation
```

Branch SHAFT_CHECK(continued)

```
SHF_INPUT_CHK                 check that shaft data are in valid ranges
|
-----GET_EXTL_CHK             check that external loads are in valid range
|
-----GET_EXTQ_CHK             check that external torques are in valid range
|
-----GET_JNLS_CHK             check that shaft segment data are in valid ranges
|
-----GET_STRS_CHK             check that stressed section data are in valid range
|
-----GET_BRGS_CHK             check that bearing data are in valid ranges
|
-----GET_KEYS_CHK             check that key data are in valid ranges

SHF_AXL_CHK            main program for shaft axial dimension consistency check
|
-----JNL_AXL_CHK              shaft segment axial dimension consistency check
|
-----GEAR_AXL_CHK             gear axial dimension consistency check
|
-----BRG_AXL_CHK              bearing axial dimension consistency check
|
-----EXT_AXL_CHK              external load axial position consistency check
|
-----TQ_AXL_CHK               external torque axial position consistency check
|
-----KEY_AXL_CHK              key axial dimension consistency check
|
-----SEC_AXL_CHK              stressed section axial position consistency check

SAVE_USER                     save user data file
|
-----OUT_FILE                 user data file name for output data(save data)
|
-----SAVE_DATA                save data file

APPS_INPUT                    application data input for shafts
|
-----GET_PW                   input power(kw or hp)

GEAR_INPUT                    gear data input
|
|----GEAR_EXTRA               extra gear data input after copying existing gear data
|    |
|    -----RPM_HH_VCT          init of rpm direction, hand of helix, centre vector
|    |
|    |----GET_VECTOR          input centre vector
|    |
|    |----GET_GRPOS           input gear axial position
|    |
|    |----GET_HELIX           input hand of helix
|
|----|
     -----RPM_HH_VCT          init of rpm direction, hand of helix, centre vector
     |
     -----GET_CENTRES_CHK     check that centre distances are in valid range
     |
     -----GET_FACES_CHK       check that face widths are in valid range
     |
     |----GET_GRTYPE          input gear types(spur, helical, double helical)
     |
     |----GET_GRPAR           input gear parameters
     |
     |----GET_CENTRES         input centre distances
     |
     |----GET_FACES           input face widths
     |
     |    |----GET_X1X2       input addendum modification coefficients
     |----|
     |    |----GET_VECTOR     input centre vector
     |
     |    |----GET_GMATER     input gear materials
     |----|
          |----GET_GRPOS      input gear axial position
```

Branch SHAFT_CHECK(continued)

```
SHF_INPUT                    main program for shaft data input
|
-----SHF_LIST                list of shaft names
|
-----SHF_SPEC                shaft data input
     |
     -----SHF_INIT(...)      individual shaft initialisation
     |
     |----GET_BRGS           input bearing data
     |    |
     |    -----BRG_FIND      find bearing designation by bearing dimensions
     |         |
     |         |----SRB_FIND for skf spherical roller bearing
     |         |
     |         |----TDI_FIND (DUMMY) for timken taper roller bearing tdi
     |         |
     |         |----TDO_FIND(DUMMY)  for timken taper roller bearing tdo
     |
     |----GET_EXTL           input external loads
     |
     |----GET_EXTQ           input external torques
     |
     |----GET_JNLS           input shaft segment/journal data
     |    |
     |    -----SHF_INIT(...) individual shaft initialisation
     |    |
     |    -----INS_JNLS      insert a segment/journal
     |    |
     |    -----DEL_JNLS      delete a segment/journal
     |    |
     |    -----GET_SKMATER   input shaft and key materials
     |         |
     |         |----I_SMATER index of shaft material
     |         |
     |         |----I_KMATER index of key material
     |         |
     |         -----INPUT_TAB select an option from a list
     |
     |----GET_STRS           input stressed section data
     |    |
     |    -----SHF_INIT(...) individual shaft initialisation
     |
     |----GET_KEYS           input key data
          |
          -----SHF_INIT(...) individual shaft initialisation
          |
          -----KEY_BHL       initialise key dimensions by shaft diameter
          |
          -----GET_SKMATER   input shaft and key materials
               |
               |----I_SMATER index of shaft material
               |
               |----I_KMATER index of key material
               |
               -----INPUT_TAB select an option from a list
```

Branch SHAFT_CHECK(continued)

```
CHECK_SHAFT                  main program for shaft check calculations
|
-----SAVE_TEMP               save temporary data file
|    |
|    -----SAVE_DATA          save data file
|
-----SHF_LIST                list of shaft names
|
|----INP_CHK(...)            input shaft check
|
|----INT_CHK(...)            intermediate shaft check
|
|----OUT_CHK(...)            output shaft check
|
|----DRVG_CHK(...)           mill pinion driving shaft check
|
|----DRVN_CHK(...)           mill pinion driven shaft check
|
-----BRG_BORE_CHK            bearing bore and gear root diameter compatibility check
|
-----BRG_OD_CHK              bearing o/d interference check
|
-----WHL_OD_CHK              wheel o/d interference check


SHF_LOAD                     shaft reactions and deflections
|
-----BEAMR_PRE               pre-treatment for beamr_fem
|
-----BEAMR_FEM               multi-support beam reaction by finite element method
|
-----LOC_POS                 locate position index on a multi-segment shaft

BRG_LIFE                     bearing life calculation
|
-----FIX_BRG                 determine which bearing as fixed
|
-----RDL_AXL                 radial and axial load for bearing
|
------
     |
     |----SRB_LIFE           skf spherical roller bearing life
     |
     |----TDI_LIFE(DUMMY)    timken taper roller tdi bearing life
     |
     |----TDO_LIFE(DUMMY)    timken taper roller tdo bearing life

BRGS_JNL                     bearing journal dimensions
|
-----LOC_POS                 locate position index on a multi-segment shaft

WHEEL                        fabricated wheel dimensions
|
-----LOC_POS                 locate position index on a multi-segment shaft
|
-----EBGD                    fabricated wheel dimensions(per bs 436-1940)
     |
     -----WMI                weight and gd2 of fabricated wheel
```

Branch SHAFT_CHECK(continued)

```
UTL_STRS                        stresses and utilisation ratios
|
-----BND_KT                     stress concentration factor for bending
|    |
|    -----LOC_POS               locate position index on a multi-segment shaft
|    |
|    -----KT_ALL                stress factors for bending, torsion and tension
|
-----TSN_KT                     stress concentration factor for torsion
|    |
|    -----LOC_POS               locate position index on a multi-segment shaft
|    |
|    -----KT_ALL                stress factors for bending, torsion and tension
|
-----LOC_POS                    locate position index on a multi-segment shaft
|
-----SEC_TQ                     torque at a section
|
-----SEC_MNT                    torque at a section

INP_CHK                         input shaft design
|
-----PPP                        pinion pitch point coordinates
|
-----SHAFT_WT                   shaft weight and gd2
|
-----SHF_LOAD(...)              shaft reactions and deflections
|
-----BRG_LIFE(...)              bearing life calculation
|
-----BRGS_JNL(...)              bearing journal dimensions
|
-----UTL_STRS(...)              stresses and utilisation ratios
|
-----SAVE_TEMP                  save temporary data file
     |
     -----SAVE_DATA             save data file

INT_CHK                         intermediate shaft check
|
-----PPP                        pinion pitch point coordinates
|
-----WPP                        wheel pitch point coordinates
|
-----SHAFT_WT                   shaft weight and gd2
|
-----WHEEL(...)                 fabricated wheel dimensions
|
-----SHF_LOAD(...)              shaft reactions and deflections
|
-----BRG_LIFE(...)              bearing life calculation
|
-----BRGS_JNL(...)              bearing journal dimensions
|
-----UTL_STRS(...)              stresses and utilisation ratios
|
-----SAVE_TEMP                  save temporary data file
     |
     -----SAVE_DATA             save data file
```

Branch SHAFT_CHECK(continued)

```
OUT_CHK                        output shaft check
|
-----WPP                       wheel pitch point coordinates
|
-----SHAFT_WT                  shaft weight and gd2
|
-----WHEEL(...)                fabricated wheel dimensions
|
-----SHF_LOAD(...)             shaft reactions and deflections
|
-----BRG_LIFE(...)             bearing life calculation
|
-----BRGS_JNL(...)             bearing journal dimensions
|
-----UTL_STRS(...)             stresses and utilisation ratios
|
-----SAVE_TEMP                 save temporary data file
     |
     -----SAVE_DATA            save data file


DRVG_CHK                       mill pinion driving shaft check
|
-----PPP                       pinion pitch point coordinates
|
-----SHAFT_WT                  shaft weight and gd2
|
-----SHF_LOAD(...)             shaft reactions and deflections
|
-----BRG_LIFE(...)             bearing life calculation
|
-----BRGS_JNL(...)             bearing journal dimensions
|
-----UTL_STRS(...)             stresses and utilisation ratios
|
-----SAVE_TEMP                 save temporary data file
     |
     -----SAVE_DATA            save data file

DRVN_CHK                       mill pinion driven shaft check
|
-----WPP                       wheel pitch point coordinates
|
-----SHAFT_WT                  shaft weight and gd2
|
-----SHF_LOAD(...)             shaft reactions and deflections
|
-----BRG_LIFE(...)             bearing life calculation
|
-----BRGS_JNL(...)             bearing journal dimensions
|
-----UTL_STRS(...)             stresses and utilisation ratios
|
-----SAVE_TEMP                 save temporary data file
     |
     -----SAVE_DATA            save data file
```

Branch SHAFT_CHECK(continued)

```
SHFC_RPT                       main program for shaft check result repor
|
-----SHF_LIST                  list of shaft names
|
-----RPT_FILE                  report file name
|
-----HEADING                   heading on report file
|
|----SHFC_OUT                  report individual shaft check results
|    |
|    -----SHF_LIST             list of shaft names
|    |
|    -----OUT_EXTQ             output external torques
|    |
|    -----OUT_JNLS             output shafts segment data
|    |    |
|    |    -----ZPART           axial position of part on shaft
|    |    |
|    |    -----PART_POS        print part name on shaft at its axial position
|    |    |
|    |    -----JNLR_POS        print shaft segment radius at its axial position
|    |
|    -----OUT_KEYD             output key dimensions
|    |
|    -----OUT_GPPS             output gear pitch point data
|    |
|    -----OUT_XYZG             output gear load components
|    |
|    -----OUT_EXTL             output external loads
|    |
|    -----OUT_BRGL             output bearing loads
|    |
|    -----OUT_BRGD             output bearing dimensions
|    |
|    -----OUT_BRGC             output bearing ratings
|    |
|    -----OUT_USTR             output utilisation ratios and stresses
|    |
|    -----OUT_KT               output stress concentration factors
|    |
|    -----OUT_DFLXN            output deflections over gear faces
|    |
|    -----OUT_ALLOW            output material allowables
|    |
|    -----OUT_WHLD             output gear wheel dimensions
|    |
|    -----OUT_WHLW             output gear wheel weights and gd2's
|
|----REPT_MESG                 report warning and diagnostic messages
|
|----PRT_FILE                  print file name
|    |
|    --------MSG_FILE          message file name
|
-----PRINTER                   printer control program
     |
     |----LASER                laser printer driver
     |
     |----LOCAL                local printer driver
          |
          -----NLEN            actual length of a character string
```

Appendix E Model of Data Organisation

```
SPACE ALLOCATION:
LINES 001 ... 100 : APPLICATION DATA
LINES 101 ... 200 : 1ST REDUCTION DATA
LINES 201 ... 300 : 2ND REDUCTION DATA
LINES 301 ... 400 : 3RD REDUCTION DATA
LINES 401 ... 500 : 4TH REDUCTION DATA
LINES 501 ... 700 : 1ST SHAFT DATA
LINES 701 ... 900 : 2ND SHAFT DATA
LINES 901 ...1100 : 3RD SHAFT DATA
LINES 1101...1300 : 4TH SHAFT DATA
LINES 1301...1500 : 5TH SHAFT DATA


                      BOX LEVEL CODES
 1 1st level calc. history code       2nd level calc. history code
 2 action code                        undo mask code
   Note:
   . 1st level calc. history code is reserved for future use;
   . 2nd level calc. history code and undo mask code bit representation:
     6 & : 15  14  13  12  11       6       5       4      3  <=BIT POSITION
     --Individual shaft done--    Shaft   Gear    Shaft  Gear<=CALCULATION
     4 & : 25  24  23  22  21     design  design  check  rating
     each bit is set to 1 for calc. done, or set to 0 to undo calc;
   . action code has a value of the above bit position for the calc.'s.
                      BOX SPECIFICATION
11 drive type(1:red'n;2:pinns;3:comp) rpm of input shaft
12 power(hp)                          power unit(1/0.7457 for hp/kw)
                      GEAR SPECIFICATION
13 overall centres                    overall centres specified
14 overall gear ratio                 number of reductions
15 contact service factor(required)   bending service factor(required)
16 power ratio for gear meshing
                      SHAFT SPECIFICATION
16                                    power ratio for bearing selection
17 power ratio for shaft design       required bearing life
18 shaft fatigue service factor       key fatigue service factor
19 weight considered in reactions(1/0) gravity direction(rad:x-y,+/-360:z)
                      DESIGN BOUNDARY VALUES
21 min single helix angle             max single helix angle
22 min double helix angle             max double helix angle
23 min overlap ratio                  max gear ratio error
24 min teeth number                   max face
25 min face to pcd ratio              max face to pcd ratio
26 min top land ratio                 max wheel pcd to face ratio
                      STANDARD DATA
29 standard to use(323/6005/2001)     normal pressure angle
30 number of grind modules            number of hob modules
31 grind module 1                     hob module 1
32 grind module 2                     hob module 2
.    .                                    .
40 grind module 10                    hob module 10
.    .                                    .
49 grind module 19                    hob module 19
50 grind module 20                    hob module 20
51 grind std specific addendum        hob std specific addendum
52 grind std specific dedendum        hob std specific dedendum
53 grind std specific tip radius      hob std specific tip radius
                      MACHINING DATA
54 protuberance, constant part        protuberance, linear coefficient
55 grind allowance, constant part     grind allowance, linear coeff.
56 grind surface finish               hob surface finish
57 min backlash, constant part        min backlash, linear coefficient
58 max backlash, constant part        max backlash, linear coefficient
```

PRECISION DATA

59	AGMA quality number(6-11)	box precision number(1-4)
60	lead modified(1/0)	gearing adjusted at assembly(1/0)
61	gears lapped(1/0)	

KEYWAY STRESS CONCENTRATION

71	keyway bending	keyway torsion

FOR FUTURE USE

72 for future use
. for future use
99 for future use

REDUCTION LEVEL CODE

101 factor specification code

Note:

factor specification code bit representation:

12	11	10	9	8	7	6	5	4	3	2	1	0	<=BIT POSITION
ZVCT	x2	x1	KR	CR	KL2	KL1	CL2	CL1	KV	CV	KM	CM	<=FACTORS

TRANSMISSION CONDITION

110	gear mesh power ratio	gear mesh power
111	gear ratio(required)	gear ratio(actual)
112	gear ratio specified(1/0)	gear ratio error(%)
113	overlap ratio	transverse contact ratio
114	rpm of the pinion(actual)	gearing life(hours)
115	contacts per flank/rev., pinion	contacts per flank/rev., wheel
116	bendings/rev., pinion	bendings/rev., wheel
117	pinion reverse/unidirection(1/0)	wheel reverse/unidirection(1/0)
118	pinion offset/bearing span, ratio	tip load(1/0)

GEAR PARTICULARS

121	spur/single/double(0/1/2)	hand of helix,pinion(-1:R,0:S,+1:L)
122	centres specified(1/0)	face specified(1/0)
123	centre	face
124	module	face to pinion pcd ratio
125	pinion teeth	wheel teeth
126	normal pressure angle	helix angle
127	specific addendum	specific dedendum
128	add. mod. coef., pinion	add. mod. coef., wheel
129	specific tip radius	

GEAR PHYSICAL SIZE

131	reference centre	
132	reference diameter,pinion	reference diameter,wheel
133	pitch diameter, pinion	pitch diameter, wheel
134	tip diameter, pinion	tip diameter, wheel
135	root diameter, pinion	root diameter, wheel

GEAR WHEEL CENTRE VECTOR

138	directional angle	z coordinate(axial position)
139	x coordinate	y coordinate

MATERIAL DATA

141	contact allowable(lb/in^2),pinion	contact allowable(lb/in^2),wheel
142	bending allowable(lb/in^2),pinion	bending allowable(lb/in^2),wheel
143	static allowable(lb/in^2), pinion	static allowable(lb/in^2), wheel
144	pinion hardness	wheel hardness
145	minimum contact allowable(<=CRS)	minimum bending allowable(<=CRS)

MANUFACTURING DATA

151	ground gears(1/0)	pinion surface finish
152	min normal backlash, pinion	min normal backlash, wheel
153	max normal backlash, pinion	max normal backlash, wheel
154	GAPS, pinion	GAPS, wheel(Grinding Allowance)
155	hob normal tooth thickness,pinion	hob normal tooth thickness,wheel
156	hob addendum(gear dedendum), pinn	hob addendum(gear dedendum), whl
157	hob dedendum(gear addendum), pinn	hob dedendum(gear addendum), whl
158	hob tip edge radius, pinion	hob tip edge radius, wheel
159	hob protuberance, pinion	hob protuberance, wheel

POWER AND STRESS RATINGS

161	pinion torque	tangential force
162	separating force	axial force
163	contact rating(hp), pinion	contact rating(hp), wheel
164	bending rating(hp), pinion	bending rating(hp), wheel
165	contact stress(lb/in^2)	
166	bending stress(lb/in^2), pinion	bending stress(lb/in^2), wheel
167	static stress(lb/in^2), pinion	static stress(lb/in^2), wheel

RATED FACTORS

170	contact service(actual), pinion	contact service(actual), wheel
171	bending service(actual), pinion	bending service(actual), wheel
172	contact safety factor, pinion	contact safety factor, wheel
173	bending safety factor, pinion	bending safety factor, wheel
174	static safety factor, pinion	static safety factor, wheel

RATING FACTORS

Note: application factor Ca and Ka is stored in service factor C_SF and K_SF

177	load distr. factor Cm	load distr. factor Km
178	dynamic factor Cv	dynamic factor Kv

179	bending geometry factor J, pinion	bending geometry factor J, wheel
180	contact geometry factor I	load sharing ratio m_N
181	stress correction factor K_f, pin	stress correction factor K_f, whl
	PSEUDO-FIXED FACTORS	
182	life factor C_L, pinion	life factor C_L, wheel
183	life factor K_L, pinion	life factor K_L, wheel
184	reliability factor C_R	reliability factor K_R
185	hardness ratio factor C_H(wheel)	rim thickness factor K_B(wheel)
186	temperature factor C_T	temperature factor K_T
187	size factor Cs, pinion	size factor Cs, wheel
188	size factor Ks, pinion	size factor Ks, wheel
189	surface condition factor Cf	elastic coefficient Cp
190	life curve class, contact(0-1)	life curve class, bending(0-1)
191	reliability, contact	reliability, bending
	FABRICATED WHEEL DATA	
192	boss outside diameter, max	boss outside diameter, min
193	rim inside diameter, max	rim inside diameter, min
194	clocking reference diameter	tube p.c.d.
195	tube size	number of tubes
196	web thickness	number of strengtheners
197	web setback	
	FABRICATED WHEEL WEIGHT	
197		weight of webs
198	weight of boss	weight of boss casting
199	weight of rim	weight of rim casting
200	wheel weight	wheel GD2

SHAFT LEVEL CODES

501 kt, tq and z specification code — journal specification code

Note:

. bit representation of kt,tq and z specification code:

```
15 14 13 12 11 10  9  8  7  6  5  4  3  2  1  0 <=BIT POSITION
   --FBRG-- ------KT_TSN-----  -----KT_BND-----
31 30 29 28 27 26 25 24 23 22 21 20 19 18 17 16 <=BIT POSITION
--TQ- -ZTQ- -------ZSEC------ --ZBRG-- --ZKEY--
```

. bit representation of journal specification code:

```
15 14 13 12 11 10  9  8  7  6  5  4  3  2  1  0 <=BIT POSITION
------------------JNL_OD-------------------- MIN
31 30 29 28 27 26 25 24 23 22 21 20 19 18 17 16 <=BIT POSITION
-------------------JNL_POS---------------------
```

SHAFT DEFINITION NUMBERS

510	number of gears(0-3)		number of bearings(2/3)	
511	number of torques(0-2)		number of keyed journals	
512	number of external loads(0-2)		number of keys	no.1
513	number of journals(1-15)		number of keys	no.2
514	number of stressed sections(0-6)		number of keys	no.3

SHAFT DESIGN DATA

515	rpm of shaft		rpm direction: (+/-)1 or 2	
516	u ratio required for shaft		u ratio required for 1 key	
517			u ratio required for 2 key	

GEAR PITCH POINTS

521	axial position	no.1	pressure angle	no.1
522	axial position	no.2	pressure angle	no.2
523	axial position	no.3	pressure angle	no.3
524	radial position	no.1	helix angle	no.1
525	radial position	no.2	helix angle	no.2
526	radial position	no.3	helix angle	no.3
527	angular position	no.1	drive mode(+1:driven)	no.1
528	angular position	no.2	drive mode(+1:driven)	no.2
529	angular position	no.3	drive mode(+1:driven)	no.3

GEAR LOADS(FOR BEARING CAPACITY)

531	x component	no.1(forward)	x component	no.1(reverse)
532	x component	no.2(forward)	x component	no.2(reverse)
533	x component	no.3(forward)	x component	no.3(reverse)
534	y component	no.1(forward)	y component	no.1(reverse)
535	y component	no.2(forward)	y component	no.2(reverse)
536	y component	no.3(forward)	y component	no.3(reverse)
537	z component	no.1(forward)	z component	no.1(reverse)
538	z component	no.2(forward)	z component	no.2(reverse)
539	z component	no.3(forward)	z component	no.3(reverse)

GEAR LOADS(FOR SHAFT STRESS)

541	x component	no.1(forward)	x component	no.1(reverse)
542	x component	no.2(forward)	x component	no.2(reverse)
543	x component	no.3(forward)	x component	no.3(reverse)
544	y component	no.1(forward)	y component	no.1(reverse)
545	y component	no.2(forward)	y component	no.2(reverse)
546	y component	no.3(forward)	y component	no.3(reverse)
547	z component	no.1(forward)	z component	no.1(reverse)
548	z component	no.2(forward)	z component	no.2(reverse)
549	z component	no.3(forward)	z component	no.3(reverse)

GEAR POWERS

551	mesh power for bearing, gear 1		mesh power for shaft, gear 1	
552	mesh power for bearing, gear 2		mesh power for shaft, gear 2	
553	mesh power for bearing, gear 3		mesh power for shaft, gear 3	

EXTERNAL TORQUES

561	torque(+:x-y)	no.1	z coordinate	no.1
562	torque(+:x-y)	no.2	z coordinate	no.2

EXTERNAL LOADS

564	x component	no.1	z coordinate	no.1
565	x component	no.2	z coordinate	no.2
566	y component	no.1	load radius	no.1
567	y component	no.2	load radius	no.2
568	z component	no.1	directional angle	no.1
569	z component	no.2	directional angle	no.2

SHAFT JOURNALS

570	minimum OD of shaft journals		start of journal	no.1
571	OD of journal	no.1	end of journal	no.1
572	OD of journal	no.2	end of journal	no.2
573	OD of journal	no.3	end of journal	no.3
574	OD of journal	no.4	end of journal	no.4
575	OD of journal	no.5	end of journal	no.5
576	OD of journal	no.6	end of journal	no.6
577	OD of journal	no.7	end of journal	no.7
578	OD of journal	no.8	end of journal	no.8
579	OD of journal	no.9	end of journal	no.9
580	OD of journal	no.10	end of journal	no.10
581	OD of journal	no.11	end of journal	no.11
582	OD of journal	no.12	end of journal	no.12
583	OD of journal	no.13	end of journal	no.13
584	OD of journal	no.14	end of journal	no.14
585	OD of journal	no.15	end of journal	no.15

SHAFT WEIGHT and GD2

586	weight of shaft		GD2 of shaft	

JOURNAL RADII

591	journal radius	no.1&2	journal radius	no.8&9
592	journal radius	no.2&3	journal radius	no.9&10
593	journal radius	no.3&4	journal radius	no.10&11
594	journal radius	no.4&5	journal radius	no.11&12
595	journal radius	no.5&6	journal radius	no.12&13
596	journal radius	no.6&7	journal radius	no.13&14
597	journal radius	no.7&8	journal radius	no.14&15
598	pinion integral on shaft			

STRESSED SECTION POSITIONS

601	position of section A		position of section D	
602	position of section B		position of section E	
603	position of section C		position of section F	

KEY DATA

604	key width	no.1	key length	no.1
605	key width	no.2	key length	no.2
606	key width	no.3	key length	no.3
607	key thickness	no.1	key ct position	no.1
608	key thickness	no.2	key ct position	no.2
609	key thickness	no.3	key ct position	no.3

BEARING SPECIFICATION

610	brg type(1/2/...)	no.1	brg selection(1/0)	no.1
611	brg type(1/2/...)	no.2	brg selection(1/0)	no.2
612	brg type(1/2/...)	no.3	brg selection(1/0)	no.3
613	brg fixed(1:Y,-1:N,0:auto)	no.1	brg min bore	no.1
614	brg fixed(1:Y,-1:N,0:auto)	no.2	brg min bore	no.2
615	brg fixed(1:Y,-1:N,0:auto)	no.3	brg min bore	no.3

BEARING DIMENSIONS

616	brg bore	no.1	brg OD	no.1
617	brg bore	no.2	brg OD	no.2
618	brg bore	no.3	brg OD	no.3
619	brg bore width	no.1	brg OD width	no.1
620	brg bore width	no.2	brg OD width	no.2
621	brg bore width	no.3	brg OD width	no.3
622	min bore abutment	no.1	max bore abutment	no.1
623	min bore abutment	no.2	max bore abutment	no.2
624	min bore abutment	no.3	max bore abutment	no.3
625	abutment radius	no.1	brg mass	no.1
626	abutment radius	no.2	brg mass	no.2
627	abutment radius	no.3	brg mass	no.3

BEARING POSITION

628	z coordinate	no.1		
629	z coordinate	no.2		
630	z coordinate	no.3		

		BEARING LOADS(FOR BEARING CAPACITY)		
631	radial load	no.1(forward)	radial load	no.1(reverse)
632	radial load	no.2(forward)	radial load	no.2(reverse)
633	radial load	no.3(forward)	radial load	no.3(reverse)
634	load angle	no.1(forward)	load angle	no.1(reverse)
635	load angle	no.2(forward)	load angle	no.2(reverse)
636	load angle	no.3(forward)	load angle	no.3(reverse)
637	axial load	no.1(forward)	axial load	no.1(reverse)
638	axial load	no.2(forward)	axial load	no.2(reverse)
639	axial load	no.3(forward)	axial load	no.3(reverse)
		BEARING LOADS(FOR SHAFT STRESS)		
641	radial load	no.1(forward)	radial load	no.1(reverse)
642	radial load	no.2(forward)	radial load	no.2(reverse)
643	radial load	no.3(forward)	radial load	no.3(reverse)
644	load angle	no.1(forward)	load angle	no.1(reverse)
645	load angle	no.2(forward)	load angle	no.2(reverse)
646	load angle	no.3(forward)	load angle	no.3(reverse)
647	axial load	no.1(forward)	axial load	no.1(reverse)
648	axial load	no.2(forward)	axial load	no.2(reverse)
649	axial load	no.3(forward)	axial load	no.3(reverse)
		BEARING CAPACITY		
651	calculated life	no.1	brg designation	no.1
652	calculated life	no.2	brg designation	no.2
653	calculated life	no.3	brg designation	no.3
654	dynamic capacity	no.1	static capacity	no.1
655	dynamic capacity	no.2	static capacity	no.2
656	dynamic capacity	no.3	static capacity	no.3
657	limiting speed	no.1,grease	limiting speed	no.1,oil
658	limiting speed	no.2,grease	limiting speed	no.2,oil
659	limiting speed	no.3,grease	limiting speed	no.3,oil
		MATERIAL ALLOWABLES		
660	shaft bending fatigue allowable		shaft shear fatigue allowable	
661	shaft bending static allowable		shaft shear static allowable	
662	key tensile yield	no.1	key shear static	no.1
663	key tensile yield	no.2	key shear static	no.2
664	key tensile yield	no.3	key shear static	no.3
		UTILISATION RATIOS		
665	@ section A	(forward)	@ section A	(reverse)
666	@ section B	(forward)	@ section B	(reverse)
667	@ section C	(forward)	@ section C	(reverse)
668	@ section D	(forward)	@ section D	(reverse)
669	@ section E	(forward)	@ section E	(reverse)
670	@ section F	(forward)	@ section F	(reverse)
671	@ key yield	no.1	@ key shear	no.1
672	@ key yield	no.2	@ key shear	no.2
673	@ key yield	no.3	@ key shear	no.3
		KEY AND SECTION STRESSES		
674	bending @ A	(forward)	bending @ A	(reverse)
675	bending @ B	(forward)	bending @ B	(reverse)
676	bending @ C	(forward)	bending @ C	(reverse)
677	bending @ D	(forward)	bending @ D	(reverse)
678	bending @ E	(forward)	bending @ E	(reverse)
679	bending @ F	(forward)	bending @ F	(reverse)
680	torsion at A		torsion at D	
681	torsion at B		torsion at E	
682	torsion at C		torsion at F	
683	@ key yield	no.1	@ key shear	no.1
684	@ key yield	no.2	@ key shear	no.2
685	@ key yield	no.3	@ key shear	no.3
		STRESS CONCENTRATION FACTORS		
686	bending factor @ A		torsion factor @ A	
687	bending factor @ B		torsion factor @ B	
688	bending factor @ C		torsion factor @ C	
689	bending factor @ D		torsion factor @ D	
690	bending factor @ E		torsion factor @ E	
691	bending factor @ F		torsion factor @ F	

DEFLECTIONS

692	@ gear 1(forward),middle-face/2	@ gear 1(reverse),middle-face/2
693	@ gear 2(forward),middle-face/2	@ gear 2(reverse),middle-face/2
694	@ gear 3(forward),middle-face/2	@ gear 3(reverse),middle-face/2
695	@ gear 1(forward),middle	@ gear 1(reverse),middle
696	@ gear 2(forward),middle	@ gear 2(reverse),middle
698	@ gear 3(forward),middle	@ gear 3(reverse),middle
698	@ gear 1(forward),middle+face/2	@ gear 1(reverse),middle+face/2
699	@ gear 2(forward),middle+face/2	@ gear 2(reverse),middle+face/2
700	@ gear 3(forward),middle+face/2	@ gear 3(reverse),middle+face/2

Appendix F Definition of COMMON Blocks

```
C--->               APPLICATION DATA
      COMMON /CODES/KODE_L1,KODE_L2,KODE_ACT,KODE_UNDO

      COMMON /BOX_SPEC/IDRIVE,RPM0,PW0,PWUNIT

      INTEGER CENTRES
      REAL    K_SF0
      COMMON /GEAR_SPEC/CRSA,CENTRES,RG0,NR0,C_SF0,K_SF0,GRPWR

      REAL    KEY_SF0
      COMMON /SHF_SPEC/BRGPWR,SHFPWR,BRGLF0,SHF_SF0,KEY_SF0,KWT,GDR

      INTEGER Z1_MIN
      COMMON /BND_VAL/HA_MIN_S,HA_MAX_S,HA_MIN_D,HA_MAX_D,OVLP_MIN,
     1       ERR_MAX,Z1_MIN,FW_MAX,FD_MIN,FD_MAX,TOP_MIN,DF_MAX

      REAL    MDL1(20),MDL2(20)
      COMMON /STD_DATA/ISTD,PA0,NMDL1,MDL1,NMDL2,MDL2,SADDG,SDEDG,
     1       STIPG,SADDH,SDEDH,STIPH

      REAL    PROT(2),GAPF(2),SFIN(2),BLMIN(2),BLMAX(2)
      COMMON /MACHINING/PROT,GAPF,SFIN,BLMIN,BLMAX

      INTEGER Q_V,B_Q,ADJUST
      COMMON /PRECISION/Q_V,B_Q,LEAD,ADJUST,LAPPED

      REAL    KT_BKEY,KT_TKEY
      COMMON /KT_KEYS/KT_BKEY,KT_TKEY
```

```
C--->               REDUCTION DATA
      INTEGER       KODE_FCT(4)
      COMMON /RDN_CODE/ KODE_FCT

      REAL    GRPWRI(4),GRPWI(4),RGR(4),RGA(4),ERR(4),OVLP(4),TRCR(4),
     1        RPM(4),GRLF(4),CONTACTS1(4),CONTACTS2(4),
     2        BENDINGS1(4),BENDINGS2(4),OFFSET(4)
      INTEGER RATIO(4),REVERSE1(4),REVERSE2(4),TIP_LOAD(4)
      COMMON /TRNSM/GRPWRI,GRPWI,RGR,RGA,RATIO,ERR,OVLP,TRCR,RPM,GRLF,
     1        CONTACTS1,CONTACTS2,BENDINGS1,BENDINGS2,
     2        REVERSE1,REVERSE2,OFFSET,TIP_LOAD

      INTEGER GRTYPE(4),HELIX(4),CENTRE(4),FACE(4),Z1(4),Z2(4)
      REAL    CRS(4),FW(4),MN(4),FD(4),PA(4),HA(4),
     1        SADD(4),SDED(4),X1(4),X2(4),STIP(4)
      COMMON /GEARP/GRTYPE,HELIX,CENTRE,FACE,CRS,FW,MN,FD,Z1,Z2,
     1        PA,HA,SADD,SDED,X1,X2,STIP

      REAL    CRS0(4),D10(4),D20(4),PCD1(4),PCD2(4),DA1(4),DA2(4),
     1        DF1(4),DF2(4)
      COMMON /GEARSIZE/CRS0,D10,D20,PCD1,PCD2,DA1,DA2,DF1,DF2

      REAL    THETA(4),ZVCT(4),XVCT(4),YVCT(4)
      COMMON /CVECTOR/THETA,ZVCT,XVCT,YVCT

      REAL    S_AC1(4),S_AC2(4),S_AT1(4),S_AT2(4),S_AY1(4),S_AY2(4),
     1        HDN1(4),HDN2(4),S_AC0(4),S_AT0(4)
      COMMON /MATERIAL/S_AC1,S_AC2,S_AT1,S_AT2,S_AY1,S_AY2,HDN1,HDN2,
     1        S_AC0,S_AT0

      INTEGER GROUND(4)
      REAL    SFINP(4),BLMIN1(4),BLMIN2(4),BLMAX1(4),BLMAX2(4),
     1        GAPF1(4),GAPF2(4),THK1(4),THK2(4),ADD1(4),ADD2(4),
     2        DED1(4),DED2(4),R_TIP1(4),R_TIP2(4),
     3        PROTUB1(4),PROTUB2(4)
      COMMON /MANUFACT/GROUND,SFINP,BLMIN1,BLMIN2,BLMAX1,BLMAX2,
     1        GAPF1,GAPF2,THK1,THK2,ADD1,ADD2,DED1,DED2,R_TIP1,R_TIP2,
     2        PROTUB1,PROTUB2

      REAL    TQ1(4),TANG(4),SEP(4),AXL(4),P_AC1(4),P_AC2(4),
     1        P_AT1(4),P_AT2(4),S_C(4),S_T1(4),S_T2(4),S_Y1(4),S_Y2(4)
      COMMON /RATINGS/TQ1,TANG,SEP,AXL,P_AC1,P_AC2,P_AT1,P_AT2,
     1        S_C,S_T1,S_T2,S_Y1,S_Y2

      REAL    C_SF1(4),C_SF2(4),K_SF1(4),K_SF2(4),C_FS1(4),C_FS2(4),
     1        K_FS1(4),K_FS2(4),K_SS1(4),K_SS2(4)
      COMMON /FACTORS/C_SF1,C_SF2,K_SF1,K_SF2,C_FS1,C_FS2,K_FS1,K_FS2,
     1        K_SS1,K_SS2

      REAL    C_M(4),K_M(4),C_V(4),K_V(4),GJ1(4),GJ2(4),GI(4),SHR(4),
     1        K_F1(4),K_F2(4)
      COMMON /FACTORS0/C_M,K_M,C_V,K_V,GJ1,GJ2,GI,SHR,K_F1,K_F2

      REAL    C_L1(4),C_L2(4),K_L1(4),K_L2(4),C_R(4),K_R(4),C_H(4),
     1        K_B(4),C_T(4),K_T(4),C_S1(4),C_S2(4),K_S1(4),K_S2(4),
     2        C_F(4),C_P(4),C_L_1E7(4),K_L_3E6(4),R_C(4),R_K(4)
      COMMON /FACTORS1/C_L1,C_L2,K_L1,K_L2,C_R,K_R,C_H,K_B,C_T,K_T,
     1        C_S1,C_S2,K_S1,K_S2,C_F,C_P,C_L_1E7,K_L_3E6,R_C,R_K

      INTEGER NTUBE(4),NSTRN(4)
      REAL    BOSSMAX(4),BOSSMIN(4),RIMMAX(4),RIMMIN(4),
     1        CLKD_WHL(4),PCD_TUBE(4),SZ_TUBE(4),THK_WEB(4),SB_WEB(4)
      COMMON /WHL_DAT/ BOSSMAX,BOSSMIN,RIMMAX,RIMMIN,CLKD_WHL,
     1        PCD_TUBE,SZ_TUBE,NTUBE,THK_WEB,SB_WEB,NSTRN

      REAL    WT_WEB(4),WT_BOSS0(4),WT_BOSS1(4),WT_RIM0(4),WT_RIM1(4),
     1        WT_WHL(4),GD2_WHL(4)
      COMMON /WHL_WT/ WT_WEB,WT_BOSS0,WT_BOSS1,WT_RIM0,WT_RIM1,
     1        WT_WHL,GD2_WHL
```

```
C--->                 SHAFT DATA
      INTEGER KODE_KTZ(5),KODE_JNL(5),KODE_NUM(5)
      COMMON /SHF_CODE/ KODE_KTZ,KODE_JNL,KODE_NUM

      INTEGER NGEAR(5),NTQ(5),NEXT(5),NJNL(5),NSEC(5),NBRG(5),
     1      NKEYJ(5),NKEYS(3,5)
      COMMON /SHF_NUM/ NGEAR,NTQ,NEXT,NJNL,NSEC,NBRG,NKEYJ,NKEYS

      INTEGER RPM_DIR(5)
      REAL    RPM_SHF(5),UTL_SHF(5),UTL_KEY(2,5)
      COMMON /SHF_DAT/ RPM_SHF,RPM_DIR,UTL_SHF,UTL_KEY

      INTEGER IDRV_GPP(3,5)
      REAL    ZGPP(3,5),RGPP(3,5),ANG_GPP(3,5),PA_GPP(3,5),HA_GPP(3,5)
      COMMON /GEAR_PP/ZGPP,RGPP,ANG_GPP,PA_GPP,HA_GPP,IDRV_GPP

      REAL    XG_BRG(3,2,5),YG_BRG(3,2,5),ZG_BRG(3,2,5),
     1        XG_SHF(3,2,5),YG_SHF(3,2,5),ZG_SHF(3,2,5)
      COMMON /GEAR_LOAD/XG_BRG,YG_BRG,ZG_BRG,XG_SHF,YG_SHF,ZG_SHF

      REAL    PW_BRG(3,5),PW_SHF(3,5)
      COMMON /GEAR_PW/PW_BRG,PW_SHF

      REAL    TQ_EXT(2,5),ZTQ(2,5)
      COMMON /EXT_TQ/TQ_EXT,ZTQ

      REAL    XL_EXT(2,5),YL_EXT(2,5),ZL_EXT(2,5),ZEXT(2,5),REXT(2,5),
     1        ANG_EXT(2,5)
      COMMON /EXT_LOAD/XL_EXT,YL_EXT,ZL_EXT,ZEXT,REXT,ANG_EXT

      REAL    OD_MIN(5),OD_JNL(15,5),POS_JNL(16,5),RD_JNL(14,5),
     1        ZSEC(6,5)
      INTEGER       INTG(5)
      COMMON /JOURNALS/OD_MIN,OD_JNL,POS_JNL,RD_JNL,ZSEC,INTG

      REAL    WT_SHF(5),GD2_SHF(5),WT_JNL(15,5),CR_JNL(15,5),
     1        WT_GEAR(3,5)
      COMMON /SHF_WT/ WT_SHF,GD2_SHF,WT_JNL,CR_JNL,WT_GEAR

      REAL    BKEY(3,5),LKEY(3,5),HKEY(3,5),ZKEY(3,5)
      COMMON /KEY_DAT/BKEY,LKEY,HKEY,ZKEY

      INTEGER BRG_TYPE(3,5),BRG_SEL(3,5),BRG_FIX(3,5)
      REAL    BRG_BORE(3,5)
      COMMON /BRG_SEPC/BRG_TYPE,BRG_SEL,BRG_FIX,BRG_BORE

      REAL    BRG_ID(3,5),BRG_OD(3,5),BRG_IW(3,5),BRG_OW(3,5),
     1        BRG_ABT0(3,5),BRG_ABT9(3,5),BRG_ABTR(3,5),BRG_WT(3,5)
      COMMON /BRG_DIM/BRG_ID,BRG_OD,BRG_IW,BRG_OW,BRG_ABT0,BRG_ABT9,
     1        BRG_ABTR,BRG_WT

      REAL    ZBRG(3,5)
      COMMON /BRG_POS/ZBRG

      REAL    RL_BRG(3,2,5),ANG_BRG(3,2,5),AXL_BRG(3,2,5),
     1        RL_SHF(3,2,5),ANG_SHF(3,2,5),AXL_SHF(3,2,5)
      COMMON /BRG_LOAD/RL_BRG,ANG_BRG,AXL_BRG,RL_SHF,ANG_SHF,AXL_SHF

      CHARACTER BRG_DSG(3,5)*18
      REAL    BRGLF(3,5),BRG_C(3,5),BRG_C0(3,5),SPD_GRS(3,5),
     1        SPD_OIL(3,5)
      COMMON /BRG_RATING/BRGLF,BRG_DSG,BRG_C,BRG_C0,SPD_GRS,SPD_OIL

      REAL    FAO_SHF(5),FQO_SHF(5),ST_SHF(5),SQ_SHF(5),
     1        SY_KEY(3,5),SQ_KEY(3,5)
      COMMON /SHF_KEY/ FAO_SHF,FQO_SHF,ST_SHF,SQ_SHF,SY_KEY,SQ_KEY

      REAL    UTL_SEC(6,2,5),YTL_KEY(3,5),STL_KEY(3,5)
      COMMON /UTL_RATIO/ UTL_SEC,YTL_KEY,STL_KEY
```

```
REAL    DFL_L(3,2,5),DFL_M(3,2,5),DFL_R(3,2,5)
COMMON /SHF_DFLXN/ DFL_L,DFL_M,DFL_R

REAL    BND_SHF(6,2,5),TSN_SHF(6,5),YLD_KEY(3,5),SHR_KEY(3,5)
COMMON /SHF_STRS/  BND_SHF,TSN_SHF,YLD_KEY,SHR_KEY

REAL    KT_BND(6,5),KT_TSN(6,5)
COMMON /KT_STRS/ KT_BND,KT_TSN
```

Appendix G Menu system of the DRIVES package

```
DRIVES
|1
|----REDUCTION BOX
|2
|----MILL PINIONS
|3
|----COMPACT DRIVES
|4
|----DATA MANAGEMENT
|5
|----EXIT
     |
     -----CONFIRMATION


     DESIRED ACTION FOR /REDUCTION BOX/MILL PINIONS/COMPACT DRIVES/
     |1
     |----BACK UP(TO PREVIOUS MENU)
     |2
     |----SAVE DATA FILE
     |3
     |----GEAR DESIGN
     |4
     |----SHAFT DESIGN
     |5
     |----GEAR RATING
     |6
     |----SHAFT CHECK


     DATA MANAGEMENT
     |1
     |----BACK UP
     |2
     |----INITIALISE DATA
     |3
     |----SAVE DATA FILE
     |4
     |----CLOSE DATA FILE
```

```
GEAR DESIGN MENU
|1
|----BACK UP
|2
|----SAVE DATA FILE
|3
|----CHOOSE GEAR STANDARD
|4
|----APPLICATION DATA
|5
|----REDUCTION DATA
|6
|----OPTIONAL DATA
|7
|----GEAR DESIGN
|8
|----RESULT REPORT

     SAVE DATA FILE
     |
     -----FILE NAME TO SAVE DATA TO:


     GEAR STANDARD TO USE
     |1
     |----AGMA323.01
     |2
     |----AGMA6005-B89
     |3
     |----AGMA2001-B88

     APPLICATION DATA
     |
     -----POWER REQUIREMENT, POWER RATIO FOR PINIONS, INPUT PINION RPM,
          OVERALL GEAR RATIO, NUMBER OF REDUCTIONS, SERVICE FACTORS OR
          APPLICATION FACTORS


     REDUCTION DATA MENU
     |1
     |----BACK UP
     |2
     |----SPECIFY REDUCTION RATIOS(LAST REDUCTION RATIO CALCULATED)
     |3
     |----SPECIFY GEAR TYPES(SPUR, SINGLE, DOUBLE)
     |4
     |----SPECIFY CENTRE DISTANCES
     |5
     |----SPECIFY FACE WIDTH
     |6
     |----SPECIFY CORRECTIONS
     |7
     |----SPECIFY MATERIALS
          |
          -----REVERSED BENDING, MATERIALS, STEEL GRADE, HARDNESS,
               ALLOWABLE STRESSES
```

GEAR DESIGN MENU(continued)

```
OPTIONAL DATA MENU
|1
|----BACK UP
|2
|----BOUNDARY VALUES
|    |
|    -----MINIMUM/MAXIMUM HELIX ANGLES, MINIMUM OVERLAP RATIO,
|         MAXIMUM GEAR RATIO ERROR, MINIMUM TEETH, MAXIMUM FACEWIDTH
|3
|----STANDARD VALUES(MODULES, PRESSURE ANGLES AND TOOTH PROPORTIONS)
|4
|----MACHINING VALUES
|    |
|    -----PROTUBERANCE,GRIND ALLOWANCE,SURFACE FINISH,BACKLASH
|5
|----PRECISION VALUES
|    |
|    -----BOX PRECISION CLASS, AGMA QUALITY NUMBER,
|         LEAD MODIFICATION, ASSEMBLY ADJUSTING, LAPPING
|6
|----PINION PROPORTION RATIOS
|    |
|    -----FACE TO PINION PCD RATIO, PINION CENTRE OFFSET RATIO
|7
|----COEFFICIENT AND FACTORS
    |
    -----ELASTIC COEFFICIENT, LOAD DISTRIBUTION FACTORS, DYNAMIC FACTORS,
         LIFE FACTORS, RELIABILITY FACTORS, TEMPERATURE FACTOR,
         SIZE FACTORS, SURFACE CONDITION FACTOR


RESULT REPORTING MENU
|1
|----BACK UP
|2
|----REPORTING MAIN RESULTS
|3
|----REPORTING FULL DETAILS
|4
|----REPORTING OPTIONAL DESIGNS
|5
|----REPORTING DESIGN CONSTRAINTS
|6
|----REPORTING WARNING MESSAGES
|7
|----PRINTING REPORT FILE
```

```
SHAFT DESIGN MENU
|1
|----BACK UP
|2
|----SAVE DATA FILE
|3
|----APPLICATION DATA
|4
|----GEARS PARTICULARS
|5
|----SHAFT SPECIFICATION
|6
|----SHAFT DESIGN
|7
|----RESULT REPORT

     APPLICATION DATA
     |
     |----POWER REQUIREMENT,(MILL PINION POWER RATIOS FOR BEARING AND STRESS),
          INPUT PINION RPM, DIRECTION(S) OF ROTATION, NUMBER OF REDUCTIONS,
          SHAFT STRENGTH SERVICE FACTOR, KEY STRENGTH SERVICE FACTOR,
          REQUIRED BEARING LIFE(HOURS)

     GEAR PARTICULARS
     |
     COPY GEAR DATA IF AVAILABLE ?
     |                             |
     YES                           NO
     |1                            |1
     |----BACK UP                  |----BACK UP
     |2                            |2
     |----SPECIFY CENTRE VECTORS   |----SPECIFY GEAR TYPES
     |3                            |3
     |----SPECIFY GEAR POSITIONS   |----SPECITY GEAR PARAMETERS
     |4                            |    |
     |----SPECIFY HAND OF HILIX    |    -----MODULE, NUMBERS OF TEETH,
          (IF SINGLE HELICAL)      |         PRESSURE ANGLE, HELIX ANGLE,
                                   |         HAND OF HELIX(IF SINGLE HELICAL),
                                   |         SPECIFIC ADDENDUM AND DEDENDUM
                                   |4
                                   |----SPECIFY CENTRE DISTANCES
                                   |5
                                   |----SPECIFY FACE WIDTHS
                                   |6
                                   |----SPECIFY CENTRE VECTORS
                                   |7
                                   |----SPECIFY GEAR POSITIONS
```

SHAFT DESIGN MENU(continued)

```
SHAFT SPECIFICATION
|1
|----BACK UP
|
|----SELECTION OF INDIVIDUAL SHAFT FOR SPECIFICATION
     |                 |                   |
     REDUCTION BOX     MILL PINIONS        COMPACT DRIVE
     |2                |2                  |2
     |----INPUT SHAFT  |----DRIVING PINION |----INPUT SHAFT
     |3                |3                  |3
     |----INTERMEDIATE |----DRIVEN PINION  |----INTERMEDIATE
     |4                |                   |4
     |----OUTPUT SHAFT |                   |----DRIVING OUTPUT
     |                 |                   |5
     |                 |                   |----DRIVEN OUTPUT
     =======================================
                       |
         INDIVIDUAL SHAFT SPECIFICATION


INDIVIDUAL SHAFT SPECIFICATION
|1
|----BACK UP
|2
|----BEARING DATA
|3
|----EXTERNAL LOADS
|4
|----EXTERNAL TORQUES
|5
|----SHAFT SEGMENTS
|6
|----STRESSED SECTIONS
|7
|----KEY DATA

     BEARING DATA
     |
     -----NUMBER OF BEARINGS ON SHAFT(2 OR 3), BEARING TYPE,
          AUTO OR MANUAL SELECTION, AXIAL POSITION, MINIMUM BORE,
          BEARING FIXTURE[A]

     EXTERNAL LOADS
     |
     -----NUMBER OF EXTERNAL LOADS(0 TO 2), AXIAL POSITION,
          RADIAL POSITION, ANGULAR POSITION, (X,Y,Z) LOAD COMPONENTS

     EXTERNAL TORQUES
     |
     -----NUMBER OF EXTERNAL TORQUES(0 TO 2), TORQUE, AXIAL POSITION

     SHAFT SEGMENTS
     |
     -----NUMBER OF SHAFT SEGMENTS(1 TO 15), INTEGRAL/SHELL PINION,
          MIN O/D OF SHAFT, STARTING POSITION OF SHAFT,
          DIAMETER[A] AND END POSITION OF EACH SEGMENT, SHAFT MATERIAL

     STRESSED SECTIONS
     |
     -----NUMBER OF STRESSED SECTIONS(0 TO 6), U RATIO REQUIRED,
          AXIAL POSITION, STRESS CONCENTRATION FACTORS[A]

     KEY DATA
     |
     |----NUMBER OF KEYED SEGMENTS(0 TO 3), U RATIO REQUIRED,
          NUMBER OF KEYS ON KEYED SEGMENT(1 TO 2),
          AXIAL POSITION OF KEY CENTRE, KEY MATERIAL
```

SHAFT DESIGN MENU(continued)

```
SHAFT DESIGN
|1
|----BACK UP
|
|----SELECTION OF INDIVIDUAL SHAFT FOR DESIGNING
         |                  |                    |
         REDUCTION BOX      MILL PINIONS         COMPACT DRIVE
         |2                 |2                   |2
         |----INPUT SHAFT   |----DRIVING PINION  |----INPUT SHAFT
         |3                 |3                   |3
         |----INTERMEDIATE  |----DRIVEN PINION   |----INTERMEDIATE
         |4                 |                    |4
         |----OUTPUT SHAFT  |                    |----DRIVING OUTPUT
         |                  |                    |5
         |                  |                    |----DRIVEN OUTPUT
         =======================================
                            |
             INDIVIDUAL SHAFT DESIGN


RESULT REPORT
|1
|----BACK UP
|
|----SELECTION OF INDIVIDUAL SHAFT FOR REPORTING
|        |                  |                    |
|        REDUCTION BOX      MILL PINIONS         COMPACT DRIVE
|        |2                 |2                   |2
|        |----INPUT SHAFT   |----DRIVING PINION  |----INPUT SHAFT
|        |3                 |3                   |3
|        |----INTERMEDIATE  |----DRIVEN PINION   |----INTERMEDIATE
|        |4                 |                    |4
|        |----OUTPUT SHAFT  |                    |----DRIVING OUTPUT
|        |                  |                    |5
|        |                  |                    |----DRIVEN OUTPUT
|        =======================================
|                           |
|            INDIVIDUAL SHAFT REPORT
|5/4/6
|----WARNING MESSAGES
|6/5/7
|----PRINT REPORT FILE
```

```
GEAR RATING MENU
|1
|----BACK UP
|2
|----SAVE DATA FILE
|3
|----CHOOSE GEAR STANDARD
|4
|----APPLICATION DATA
|5
|----GEAR PARTICULARS
|6
|----OPTIONAL DATA
|7
|----GEAR RATING
|8
|----RESULT REPORT


     SAVE DATA FILE
     |
     -----FILE NAME TO SAVE DATA TO:


     GEAR STANDARD TO USE
     |1
     |----AGMA323.01
     |2
     |----AGMA6005-B89
     |3
     |----AGMA2001-B88


     APPLICATION DATA
     |
     -----POWER REQUIREMENT, POWER RATIO FOR PINIONS, INPUT PINION RPM,
          OVERALL GEAR RATIO, NUMBER OF REDUCTIONS, SERVICE FACTORS OR
          APPLICATION FACTORS


     GEAR PARTICULARS
     |1
     |----BACK UP
     |2
     |----SPECIFY GEAR TYPES
     |3
     |----SPECITY GEAR PARAMETERS
     |    |
     |    -----MODULE, NUMBERS OF TEETH OF PINION AND WHEEL
     |4
     |----SPECIFY CENTRE DISTANCES
     |5
     |----SPECIFY FACE WIDTHS
     |6
     |----SPECIFY CORRECTIONS
     |7
     |----SPECIFY MATERIALS
          |
          -----REVERSED BENDING, MATERIALS, STEEL GRADE, HARDNESS,
               ALLOWABLE STRESSES
```

GEAR RATING MENU(continued)

```
OPTIONAL DATA MENU
|1
|----BACK UP
|2
|----BOUNDARY VALUES
|     |
|     -----MINIMUM/MAXIMUM HELIX ANGLES, MINIMUM OVERLAP RATIO,
|          MAXIMUM GEAR RATIO ERROR, MINIMUM TEETH, MAXIMUM FACEWIDTH
|3
|----STANDARD VALUES(MODULES, PRESSURE ANGLES AND TOOTH PROPORTIONS)
|4
|----MACHINING VALUES
|     |
|     -----PROTUBERANCE,GRIND ALLOWANCE,SURFACE FINISH,BACKLASH
|5
|----PRECISION VALUES
|     |
|     -----BOX PRECISION CLASS, AGMA QUALITY NUMBER,
|          LEAD MODIFICATION, ASSEMBLY ADJUSTING, LAPPING
|6
|----PINION PROPORTION RATIOS
|     |
|     -----FACE TO PINION PCD RATIO, PINION CENTRE OFFSET RATIO
|7
|----COEFFICIENT AND FACTORS
    |
    -----ELASTIC COEFFICIENT, LOAD DISTRIBUTION FACTORS, DYNAMIC FACTORS,
         LIFE FACTORS, RELIABILITY FACTORS, TEMPERATURE FACTOR,
         SIZE FACTORS, SURFACE CONDITION FACTOR


RESULT REPORTING MENU
|1
|----BACK UP
|2
|----REPORTING MAIN RESULTS
|3
|----REPORTING FULL DETAILS
|4
|----REPORTING WARNING MESSAGES
|5
|----PRINTING REPORT FILE
```

```
SHAFT CHECK MENU
|1
|----BACK UP
|2
|----SAVE DATA FILE
|3
|----APPLICATION DATA
|4
|----GEAR PARTICULARS
|5
|----SHAFT SPECIFICATION
|6
|----SHAFT CHECK
|7
|----RESULT REPORT

     SAVE DATA FILE
     |
     -----FILE NAME TO SAVE DATA TO:

     APPLICATION DATA
     |
     |----POWER REQUIREMENT,(MILL PINION POWER RATIOS FOR BEARING AND STRESS),
          INPUT PINION RPM, DIRECTION(S) OF ROTATION, NUMBER OF REDUCTIONS,
          SHAFT STRENGTH SERVICE FACTOR, KEY STRENGTH SERVICE FACTOR,
          REQUIRED BEARING LIFE(HOURS)

     GEAR PARTICULARS
     |
     COPY GEAR DATA IF AVAILABLE ?
     |                             |
     YES                           NO
     |1                            |1
     |----BACK UP                  |----BACK UP
     |2                            |2
     |----SPECIFY CENTRE VECTORS   |----SPECIFY GEAR TYPES
     |3                            |3
     |----SPECIFY GEAR POSITIONS   |----SPECITY GEAR PARAMETERS
     |4                            |    |
     |----SPECIFY HAND OF HILIX    |    -----MODULE, NUMBERS OF TEETH,
          (IF SINGLE HELICAL)      |         PRESSURE ANGLE, HELIX ANGLE,
                                   |         HAND OF HELIX(IF SINGLE HELICAL),
                                   |         SPECIFIC ADDENDUM AND DEDENDUM
                                   |4
                                   |----SPECIFY CENTRE DISTANCES
                                   |5
                                   |----SPECIFY FACE WIDTHS
                                   |6
                                   |----SPECIFY CENTRE VECTORS
                                   |7
                                   |----SPECIFY GEAR POSITIONS
```

SHAFT CHECK MENU(continued)

```
SHAFT SPECIFICATION
|1
|----BACK UP
|
|----SELECTION OF INDIVIDUAL SHAFT FOR SPECIFICATION
     |                  |                   |
     REDUCTION BOX      MILL PINIONS        COMPACT DRIVE
     |2                 |2                  |2
     |----INPUT SHAFT   |----DRIVING PINION |----INPUT SHAFT
     |3                 |3                  |3
     |----INTERMEDIATE  |----DRIVEN PINION  |----INTERMEDIATE
     |4                 |                   |4
     |----OUTPUT SHAFT  |                   |----DRIVING OUTPUT
     |                  |                   |5
     |                  |                   |----DRIVEN OUTPUT
     =======================================
                        |
         INDIVIDUAL SHAFT SPECIFICATION


    INDIVIDUAL SHAFT SPECIFICATION
    |1
    |----BACK UP
    |2
    |----BEARING DATA
    |3
    |----EXTERNAL LOADS
    |4
    |----EXTERNAL TORQUES
    |5
    |----SHAFT SEGMENTS
    |6
    |----STRESSED SECTIONS
    |7
    |----KEY DATA

         BEARING DATA
         |
         -----NUMBER OF BEARINGS ON SHAFT(2 OR 3), BEARING TYPE,
              AXIAL POSITION, BEARING BORE, O/D, WIDTH, BEARING FIXTURE

         EXTERNAL LOADS
         |
         -----NUMBER OF EXTERNAL LOADS(0 TO 2), AXIAL POSITION,
              RADIAL POSITION, ANGULAR POSITION, (X,Y,Z) LOAD COMPONENTS

         EXTERNAL TORQUES
         |
         -----NUMBER OF EXTERNAL TORQUES(0 TO 2), TORQUE, AXIAL POSITION

         SHAFT SEGMENTS
         |
         -----NUMBER OF SHAFT SEGMENTS(1 TO 15), INTEGRAL/SHELL PINION,
              STARTING POSITION OF SHAFT, DIAMETER AND END POSITION
              OF EACH SEGMENT, RADIUS BETWEEN SEGMENTS, SHAFT MATERIAL

         STRESSED SECTIONS
         |
         -----NUMBER OF STRESSED SECTIONS(0 TO 6), AXIAL POSITION,
              STRESS CONCENTRATION FACTORS[A]

         KEY DATA
         |
         |----NUMBER OF KEYED SEGMENTS(0 TO 3), NUMBER OF KEYS ON KEYED
              SEGMENT(1 TO 2), AXIAL POSITION OF KEY CENTRE,
              KEY DIMENSIONS INITIALISED BY SEGMENT(?), KEY WIDTH,
              KEY THICKNESS, KEY LENGTH, KEY MATERIAL
```

SHAFT CHECK MENU(continued)

```
SHAFT CHECK
|1
|----BACK UP
|
|----SELECTION OF INDIVIDUAL SHAFT FOR CHECKING
     |                  |                   |
     REDUCTION BOX      MILL PINIONS        COMPACT DRIVE
     |2                 |2                  |2
     |----INPUT SHAFT   |----DRIVING PINION |----INPUT SHAFT
     |3                 |3                  |3
     |----INTERMEDIATE  |----DRIVEN PINION  |----INTERMEDIATE
     |4                 |                   |4
     |----OUTPUT SHAFT  |                   |----DRIVING OUTPUT
     |                  |                   |5
     |                  |                   |----DRIVEN OUTPUT
     ======================================
                        |
         INDIVIDUAL SHAFT CHECK


RESULT REPORT
|1
|----BACK UP
|
|----SELECTION OF INDIVIDUAL SHAFT FOR REPORTING
|    |                  |                   |
|    REDUCTION BOX      MILL PINIONS        COMPACT DRIVE
|    |2                 |2                  |2
|    |----INPUT SHAFT   |----DRIVING PINION |----INPUT SHAFT
|    |3                 |3                  |3
|    |----INTERMEDIATE  |----DRIVEN PINION  |----INTERMEDIATE
|    |4                 |                   |4
|    |----OUTPUT SHAFT  |                   |----DRIVING OUTPUT
|    |                  |                   |5
|    |                  |                   |----DRIVEN OUTPUT
|    ======================================
|                       |
|        INDIVIDUAL SHAFT REPORT
|5/4/6
|----WARNING MESSAGES
|6/5/7
|----PRINT REPORT FILE
```

Appendix H Full Expansion of Parametric Program Structure

```
LAYOUT                  master procedure for layout drawing
|
CHK_VAR(...)            check if variables already declared
|
DCL_ALL                 declare all variables
|
READ_DATA(...)          read data file
|
INIT_DATA               initialise variables with data from file
|
FRONT(...)              create front elevation
|
OTHERS(...)             create other elevations


FRONT
|
-----FRONT_MODEL(...)        create front model cell
|
-----GET_DIRECTIONS          get directions for the coordinates
|    |
|    -----X_DIRECTION        direction for positive X
|    |
|    -----Y_DIRECTION        direction for positive Y
|    |
|    -----Z_DIRECTION        direction for positive Z
|    |
|    -----CHK_DIRECTIONS     check if right hand rule followed
|         |
|         |----NOTHING
|         |
|         |----GET_DIRECTIONS
|
-----CRS_VECTORS             establish centre vectors
|
-----SHF_CENTRES             establish shaft centres
|
-----DRAW_FRONT              draw front elevation
|    |
|    -----WIND_SCALE         determine size of the window
|
-----DIME_FRONT              dimension front elevation
     |
     -----TRI_ARROW          arrow for direction of rotation
```

```
OTHERS
|
|----ELEVATIONS(...)       create other elevations
|
|----OPEN_MODEL            list names of cells and open one
|
|----PLOT_WINDOW           plot the drawing shown on screen
|
|----EXIT(...)             shut down and exit

ELEVATIONS
|
-----OPEN_MODEL_ FRONT             open front model cell
|
|----SECTIONS                      create plane sectional view
|    |
|    |----HORIZONTAL_V             horizontal sectional view
|    |    |
|    |    -----NEW_MODEL(...)      create a new model cell
|    |    |
|    |    -----TRANSF_CRD          transformation of coordinates
|    |    |
|    |    -----WIND_SCALE          determine size of the window
|    |    |
|    |    -----DRAW_SHAFT          draw shaft
|    |    |
|    |    -----DRAW_BEARING        draw bearing
|    |    |
|    |    -----DRAW_PINION         draw pinion
|    |    |    |
|    |    |    -----LOC_POS        locate position
|    |    |
|    |    -----DRAW_WHEEL          draw wheel
|    |    |    |
|    |    |    -----LOC_POS        locate position
|    |
|    |----VERTICAL_V               vertical sectional view
|         |
|         -----(same as HORIZONTAL_V)
|
|----EXPANDED                      create expanded section view
     |
     |----HORIZONTAL               horizontal expansion
     |    |
     |    ---HORIZONTAL_X(...)     horizontal expanded view
     |
     |----VERTICAL                 vertical expansion
          |
          ---VERTICAL_X(...)       vertical expanded view
```

```
HORIZONTAL_X                 horizontal expanded view
|
-----NEW_MODEL               create a new model cell
|
-----TRANSF_CRD              transformation of coordinates
|
-----WIND_SCALE              determine size of the window
|
-----DRAW_SHAFT              draw shaft
|
-----DRAW_BEARING            draw bearing
|
-----DRAW_PINION             draw pinion
|    |
|    -----LOC_POS            locate position
|
-----DRAW_WHEEL              draw wheel
|    |
|    -----LOC_POS            locate position
|
-----DIME_HORIZONTAL         dimension for horizontal expansion
     |
     -----DIME_SETUP         dimensioning setup
     |
     -----DIME_GEAR_H        datum and main gear dimensions
     |    |
     |    -----LOC_POS       locate position
     |
     -----DIME_BRG_H         bearing spans, identification mark
     |
     -----DIME_SHF_H         keyed diameters, overall length
     |    |
     |    -----LOC_POS       locate position
     |
     -----DIME_INFO          text information
     |    |
     |    -----BRG_BOX       bearing data enveloping box
     |    |    |
     |    |    BOX_ANCHOR    anchor point for the box
     |    |
     |    -----BRG_INFO      bearing data: sizes, lives, etc.
     |    |
     |    -----WARN_BOX      warning message enveloping box
     |    |    |
     |    |    BOX_ANCHOR    anchor point for the box
     |    |
     |    -----WARN_MSG      warning message
     |
     -----DIME_MASK          text mask underlying lines
     |
     -----DIME_RESTORE       restore original dimension setup
          |
          ---TEXT_RESTORE    restore original text setup
```

```
VERTICAL_X                   vertical expanded view
|
-----NEW_MODEL               create a new model cell
|
-----TRANSF_CRD              transformation of coordinates
|
-----WIND_SCALE              determine size of the window
|
-----DRAW_SHAFT              draw shaft
|
-----DRAW_BEARING            draw bearing
|
-----DRAW_PINION             draw pinion
|     |
|     -----LOC_POS           locate position
|
-----DRAW_WHEEL              draw wheel
|     |
|     -----LOC_POS           locate position
|
-----DIME_VERTICAL           dimension for vertical expansion
      |
      -----DIME_SETUP        dimensioning setup
      |
      -----DIME_GEAR_V       datum and main gear dimensions
      |     |
      |     -----LOC_POS     locate position
      |
      -----DIME_BRG_V        bearing spans, identification mark
      |
      -----DIME_SHF_V        keyed diameters, overall length
      |     |
      |     -----LOC_POS     locate position
      |
      -----DIME_INFO         text information
      |     |
      |     -----BRG_BOX     bearing data enveloping box
      |     |     |
      |     |     BOX_ANCHOR anchor point for the box
      |     |
      |     -----BRG_INFO    bearing data: sizes, lives, etc.
      |     |
      |     -----WARN_BOX    warning message enveloping box
      |     |     |
      |     |     BOX_ANCHOR anchor point for the box
      |     |
      |     -----WARN_MSG    warning message
      |
      -----DIME_MASK         text mask underlying lines
      |
      -----DIME_RESTORE      restore original dimension setup
            |
            ---TEXT_RESTORE  restore original text setup
```

```
CHK_VAR                     check if variables already declared
|
|----NOTHING                dummy CPROC
|
|----|
     -----DIME_RESTORE      restore original dimensioning setup
     |
     -----UNDCL_ALL         undeclare all variables

READ_DATA                   read data
|
-----OPEN_FILE              open data file


FRONT_MODEL                 create front model cell
|
-----CREATE_MODEL_          create a model cell with fixed name
|
-----CHK_CELL_EMPTY         check if cell empty
      |
      |----NOTHING          dummy CPROC
      |
      |----CELL_NOT_EMPT    cell has geometry
            |
            |----DEL_ALL    delete all geometry


NEW_MODEL                   create new model cell
|
-----CREATE_MODEL           create a model cell with user name
|
-----CHK_CELL_EMPTY         check if cell empty
      |
      |----NOTHING          dummy CPROC
      |
      |----CELL_NOT_EMPT    cell has geometry
            |
            |----DEL_ALL    delete all geometry


EXIT                        shut down and exit
|
-----DIME_RESTORE           restore original dimension setup
|
-----UNDCL_ALL              undeclare all variables
```