# COMPUTER INTEGRATED MANUFACTURE OF TURNED SHAFT ASSEMBLIES

by


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B.E., Osmania University, Hyderabad, INDIA, 1986


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A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE


Department of Mechanical Engineering

KANSAS STATE UNIVERSITY

Manhattan, Kansas


1988

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ACKNOWLEDGEMENTS

I would like to thank the following people for their assistance and contributions to this thesis: Dr. Prakash Krishnaswami, whose guidance and enthusiasm made this work possible, Dr. Huang, C. L. for his guidance in dynamics concepts and for serving on my committee, Dr. Fred Appl for serving on my committee, the Department of Mechanical Engineering, Kansas State University, and my fellow students for their encouragement and support.

# CHAPTER I

## INTRODUCTION

Since the industrial revolution new and better ways of manufacturing machined components have been continually and eagerly sought after. Engineers in the past have made rapid progress in machining technology and today it is possible to produce high quality machined components in large quantities. The high volume of machined components that must be produced by the manufacturing industry has motivated engineers to seek better machining procedures so that machined components of the required precision are produced at minimum cost or at the maximum production rate.

During the machining of any component, it is first necessary to set the quality specifications like surface finish, manufacturing tolerance, accuracy, etc. While machining the component within these specifications, the machinist has a wide range of speeds, feeds, tool materials and other machining conditions that are under his control. In order to meet the preset specifications, the machinist is usually conservative in the selection of a set of machining parameters, since meeting specifications takes precedence over reducing machining cost. Unfortunately, the conventional practice followed in the manufacture and design of machined components is to judge by experience and not by mathematical reasoning. This is uneconomical in the long run. There is also a lack of good data in this area to assist the machinist in taking economical and safe decisions. Hence there is a strong need to collect information about manufacturing and design practices, and to organize this information so as to arrive at a set of machining parameters which satisfy the given specifications and provide minimum cost or minimum cutting time per piece. Also, little effort has been made in the past to link the design of components with the manufacturing cost. This is a very important concept and is one that is very helpful in attaining the objective of efficient manufacture. For example consider a single shaft supported by sleeve bearings at the ends and carrying gears or pulleys as shown in Fig. 1.1. The loads are specified but the placement of the loads is to be decided by the designer. There are two possibilities :

(i) to place the loads according to set practices which may not guarantee minimum deflection or shear stress

(ii) to find a position such that the maximum deflection and the maximum bending moment on the shaft and the corresponding values of shear stress are minimized so that the shaft can be made as small as possible.

The second approach, if interlinked with the optimization of manufacturing cost , leads to a more economical and safer solution .

The surface finish and manufacturing tolerance specified by the designer have a direct bearing on the manufactured cost of a component. A machine component turned to a better surface finish

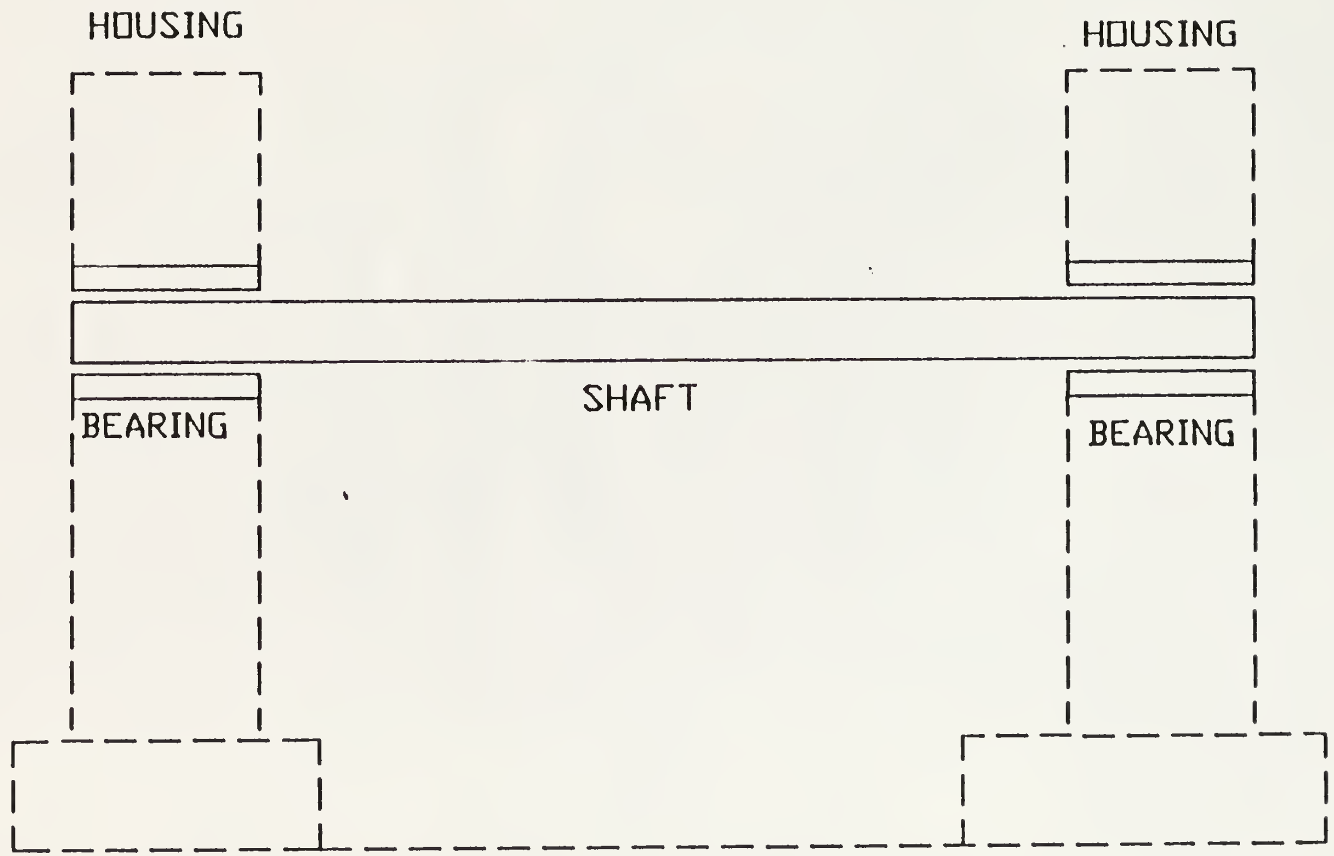



Figure 1.1

obviously costs more to produce than one with a coarse surface finish. Similarly, the dimensions of the components have a direct effect on the manufactured cost of the component. Also, if a part is to be machined to a very low tolerance, the feed must be kept low and so the machining time and the manufacturing cost are increased. Hence there is also a need to check the design process to see that the tolerances specified do not become too demanding while ensuring that they safely satisfy the desired performance requirements.

The above discussion makes it clear that in order to minimize manufacturing cost or machining time, it is necessary to relate the manufacturing requirements like surface finish, manufacturing tolerance, etc. to the cutting parameters. Furthermore, the design requirements such as limits on maximum allowable shear stress, maximum deflection, etc. must be related to the manufacturing cost/time. If such relationships can be established, it will be possible to simultaneously design the component and select the cutting parameters for its manufacture in such a way that all design requirements are satisfied and the manufacturing cost/time is minimized.

Past efforts in this area have been largely aimed at arriving at an overall expression relating tool life to cutting speed, feed and other cutting parameters and calculating the optimum cutting speed based on these relationships. Most of these models are incomplete in that they do not account for all relevant factors. Further, none of them accounts for the effect of the design on the manufactured cost.

Holmes [1] made a detailed study of machining practices and produced a large number of tables, charts and other material based upon his studies of the turning process. This work is a commendable attempt to correlate the control of manufacturing tolerance with the cutting parameters in machining practice. However, the work is somewhat crude in several aspects, such as the suggested criteria for deciding the values of constants and coefficients to be used in empirical formulas. Nevertheless, the data given in these tables is of great practical value and can be made more usable if it is entered into a computer database .

The General Electric Data System [2] was developed to demonstrate the use of empirical machinability parameters in a mathematical computer model to determine optimum machining conditions for minimum cost and maximum production. This system has been used on a commercial basis with some success.

FAST (Feed And Speed Technology) [3] is a programming system that was developed to relieve the programmer of lengthy , tedious selection of feeds and speeds , and to allow the production cost to be decreased by providing automatic selection of proper tools with correct cutting tool geometries . FAST also ensures accuracy , flexibilty and quick implementation of changes that result from the development of better tools and manufacturing methods .

The approaches presented in [2] and [3] were reasonably successful efforts to computerize machinability data; unfortunately these methods do not consider the implementation of the design requirements. Further, the models developed in these references use a fairly large number of constants whose values are set somewhat arbitrarily, based on past experience. Reasonable values for the constants are not always available, especially at nonstandard machining conditions.

Bhattacharya et. al. [4] first explored the idea of applying constraints on the manufacturing cost function to ensure satisfaction of surface finish specifications. The speeds, feeds and other cutting parameters were specified with bounds to obtain the minimum cost per piece. The satisfaction of prescribed tolerances was not considered in this work.

Hati and Rao [5] applied mathematical programming techniques to determine the cutting parameters based on three different objective functions: the minimum cost of production, the maximum production rate and the maximum total profit. Comparison of so called deterministic and probabilistic approaches was carried out. The constraints used in the model were the bounds on speed, feed and depth of cut as well as the limits on the cutting force, power and temperature encountered in the turning operation . Here again the number of constraints was very low and tolerance requirements were overlooked.

It follows from this discussion of the literature that there are two aspects that have been left largely unexplored:

(i)optimal selection of cutting parameters from a continuous range of available values rather than selecting from a small number of possibilities given in standard tables ;

(ii)optimal design of components such that the manufacturing cost/time required for producing the component is minimized while ensuring that the component meets all design specifications.

This thesis is an attempt to explore both these avenues with a view to developing an integrated approach to the optimal design and manufacture of a component. Ideally experimental work has to be done in order to derive better relationships between surface finish , manufacturing tolerance , tool nose radius , depth of cut and other machining parameters . Unfortunately suitable experimental facilities were not available and so another approach was used which took maximum advantage of published data. In this approach, approximating functions are fitted to existing tables available in the literature which relate the cutting parameters to the accuracy and surface finish of the finished work piece. In addition, expressions drawn from the literature are used to form estimates of cutting time, tool life, etc. Using these functions, it is possible to construct a mathematical model which can be used to select optimal values for the manufacturing parameters to minimize the manufacturing cost while satisfying requirements on part tolerances, surface roughness, etc.

An optimization model is developed, considering the manufacturing cost as the cost function. The various manufacturing requirements like the part tolerances, surface finish and the surface fit requirements are enforced through constraints. A standard nonlinear optimization problem can be considered in the following form :

Minimize $F(B)$

subject to the constraints

$$g_j(B) \leq 0 \ , \quad j = 1 \ , m$$

$$g_k(B) = 0 \ , \quad k = m+1, \ n \qquad 1.1$$

where $n$ is the total number of constraints,

$m$ is the number of inequality constraints, and

$B$ is the design vector which contains the various independent design variables $[b_1, \ b_2, \ b_3, \ .......b_{nv}]^T$ with nv being the number of design variables.

The problem of obtaining optimum machining parameters can be converted into a standard nonlinear programming problem of the above form.

The above optimization problem statement is broad enough to cover design considerations as well. Since the optimization problem can have any number of equality and inequality constraints , the performance requirements of the component such as limits on shear stress, maximum deflection, etc. can all be brought into this optimization problem by the introduction of additional constraints. Furthermore, the design parameters of the components ( lengths , inner and outer diameters, etc.) can be added to the set of design variables for the optimization problem. Thus, by solving a single optimization problem we can find an optimal solution to the component design problem as well as the component manufacture problem. Since the solution is done simultaneously the component design that is arrived at and the cutting parameters that are selected will be the ones associated with the minimum manufactured cost.

The aim of this work is to develop a scheme for integrating design and manufacture using the approach described above for the specific case of shaft assemblies. A secondary goal is the implementation of this scheme in a reliable optimal design code which can be used for solving problems in the design and manufacture of transmission shaft assemblies. In the first phase of this work, the design of the shaft assembly was kept fixed and a nonlinear optimization formulation was derived whereby the optimal machining parameters were determined for minimum cost of manufacture while satisfying all the manufacturing requirements. Some examples were run and the results obtained were satisfactory. The optimization scheme was then extended to integrate design and manufacture within a single optimization problem. Again several examples were run and the approach was found to work reliably and effectively. The program implementation is designed to be flexible to give the user the capability to define the design vector based upon the demands of the particular problem at hand.

In chapter II , the details regarding the formulation of a surface finish prediction model and a manufacturing tolerance prediction model are discussed. A review of the tool life equation is also presented. The cost function for manufacturing cost is developed in chapter III. A detailed discussion of the formulation of manufacturing constraints like surface finish, manufacturing tolerance and mating fit requirements is also presented. Chapter IV outlines a mathematical programming approach to the integration of component design with component manufacture. A discussion of the additional constraints needed to ensure satisfaction of design requirements is also included. Chapter V gives the details of the optimization algorithm that was used in this work. A detailed explanation of the development of a computer code based on the methods described in other chapters is also presented. Chapter VI discusses some numerical examples that were solved using the proposed approach. The results attest to the feasibility and effectiveness of the method. Finally, a brief conclusion and some recommendations for future research are presented in chapter VII .

# CHAPTER II

# MATHEMATICAL MODELS

In any mathematical programming solution to an engineering problem, mathematical models play an important part and form the basis for the prediction of system behavior. Hence, the formulation of mathematical models in machinability problems needs serious consideration. When machinability is analysed from the systems view point, electrical power, human effort, raw material, machine tools and perishable cutting tools may be considered as inputs to the system, with machined products as the output. The main operating purpose of a manufacturing business is to control the inputs in order to produce a work piece at either the minimum cost or the maximum production rate. There are definite operating conditions under which these two manufacturing objectives may be achieved. It is the goal of the engineer working with the economics of machining to predict the operating conditions that meet these objectives.

The first step in using the computer to determine optimum machining conditions is to prepare mathematical models that include all the significant parameters. There are two possible approaches in this regard. The first is to write one complete mathematical formula to account for all machining parameters simultaneously. Gilbert [6] of the General Electric Company developed the following formula for the recommended cutting speed.

$$SPEED = \frac{CONST*\ COOLF*SURF*TMATF\ *PROFF*FLANK^{.25}}{K^{1.72}*TLIFE^{n}*FEED^{.58}*DC^{.2}}$$

$$* \ (BHN_r)/(BHN_w))^{1.72}* \ [MR] \qquad 2.1$$

where :

CONST= A constant dependent upon the basic tool material

COOLF = Coolant factor based upon coolant being used

SURF = Surface Factor describing whether surface is clean , sand cast or heat treated .

TMATF = Tool material factor

PROFF = Profile factor - a function of nose radius , depth of cut , and cutting edge angle

FLANK = Flank Wear Factor

K = the Brinell hardness number for the base material, which is usually AISI B1112 steel with has a BHN of 160

TLIFE = Tool life

n = Slope of the tool-life line

FEED = Feed

DC = Depth of cut

$BHN_r$ = Brinell hardness number at which the machinability rating was established

$BHN_w$ = Brinell hardness number of the work piece

MR = Machinability rating of work piece material established at the hardness indicated.

The various constants and parameters are to be selected from combinations of single , double and triple variable graphs.

This method predicts the cutting speed by substituting all the related parameter values. However the values of most of the constants used in this model have to be chosen from graphs and tables and some of the parameters are not sufficiently well understood to be represented by continuous functions. Further, this equation does not consider part tolerance. Hence the equation is unsuitable for use in a mathematical model that is to serve as a basis for optimization . Also, the use of multiple constraints rather than a single constraint is a more reasonable way to direct the machining parameters towards an optimum solution.

In the present work, the determination of optimum machining parameters is carried out using iterative optimization techniques. Several neccessary constraints representing various design and manufacturing criteria are applied to the cost function. Among the manufacturing criteria are the surface finish requirements and the manufacturing tolerance requirements. Hence mathematical models for these two quantities must be developed .

## 2.1 Surface Finish Model

The prediction of surface finish based upon the cutting parameters is an essential component of any mathematical model for manufacturing. The prediction must be accurate and complete, taking into account as many of the relevant factors as possible. Such a model allows the user to select the cutting parameters to be used in order to arrive at a desired surface finish. In the context of mathematical programming methods for the minimization of manufacturing cost or maximization of production rate, the surface finish model may be used to formulate constraints whose satisfaction will ensure that the cutting parameters selected are consistent with the surface finish required.

Among past efforts aimed at developing a surface finish model, the work of Bhattacharya et. al. [4] and the research conducted by the General Electric Company [2,3] are the most notable. The model presented in [4] describes surface finish as a function of feed and cutting speed. The functional dependency is of the form :

$$R_a = 10^4 * (K_1 * F^{K_2}) \qquad 2.2$$

where

$R_a$ is the Center Line Average (CLA) value of surface roughness in micro in. ,

F is the feed in in. per revolution ,

$K_1$ and $K_2$ are constants depending on cutting speed, tool geometry, environment etc.

This model does not consider nose radius as one of the parameters. Also the values of $K_1$ and $K_2$ are discontinuous and cannot be used if the surface finish model is to be a continuous function over all ranges of values of the parameters.

The research conducted by General Electric has resulted in five diferent surface finish models :

(i)Gilbert's model [6] makes use of a parameter called the surface factor (SURF in Eq. 2.1). This factor does not give a clear picture of the surface finish ; rather, it is a very crude input to the model (1.0 for a clean surface, .8 for a heat treated surface, and .7 for a sand-cast surface) for obtaining an approximation of the optimum cutting speed.

(ii) The General Electric Data System [2] uses the following expression involving nose radius, theoretical surface finish and feed.

$$\mathrm{FEED} = \{(21.6 * \mathrm{RNOSE} * \mathrm{RMS})/(f_{finish})\}^{.5} * 1.0\ \mathrm{E} - 03 \qquad 2.3$$

where, RNOSE is the nose radius of the tool in inches ,

FEED is the cutting feed in ipr ,

RMS is the surface finish expressed in micro inches (root mean square)

$f_{finish}$ is the finish factor which is an empirical value that depends on the tool and job materials.

This expression indicates that surface finish is directly proportional to the square of the feed and inversely proportional to nose radius . This model does not account for the cutting speed which, as discussed later, is closely related to the surface finish. Also there is some difficulty in establishing the values of the finish factor to be used in this model, as no information is available in the literature regarding this parameter .

(iii) GE Data Systems [2] gives the dependency of surface finish in the form of a graph as shown in Fig. 2.2 .This graph gives the values of surface finish as a function of nose radius for various feed values.

(iv) The Carboloy Systems department of the General Electric Company also gives a similar graph with the same general trend [6] but gives slightly different results, as shown in Fig. 2.3 .

(v) Carboloy Systems [6] also suggest the following surface roughness model :

$$A.A. = 2.*(R^2 - H*G)/F \qquad 2.4$$

where ,

R = Nose Radius in inches

A.A. = Surface Roughness (arithmetic average method)

F = Feed Rate

H = Distance from the center of the nose radius to the mean line of the generated peak and valleys of the work piece profile ,

G = Perpendicular distance to the mean line from the normal center line of the radius to the intersection

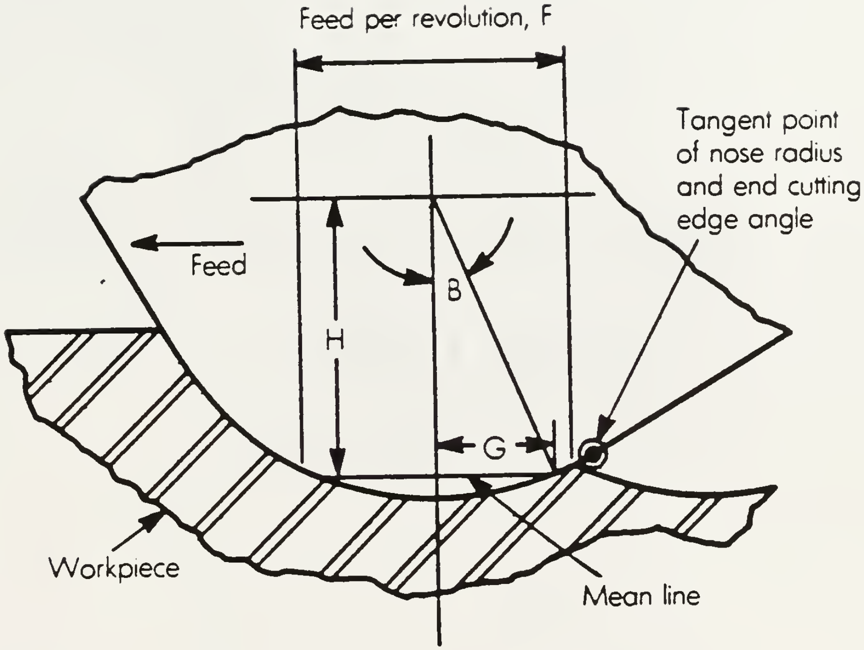



Fig. 2.1 Elements used to calculate theoretical surface roughness in single point turning and boring operations.

(Reproduced from General Electric Carboloy) of the generated profile

B = The angle between the normal center line and the radial line which intersects the mean line at the cutting point.

Fig. 2.1 shows the various parameters used in the above formula.

However this model indicates an inverse relationship with feed and a direct proportionality with the square of nose radius,i.e. as the nose radius of the tool increases the surface finish increases and as the feed increases the surface finish decreases. According to all the other models as nose radius increases the surface finish must decrease and as the feed increases the surface finish must increase. Thus model (v) contradicts all the other models and it becomes neccessary to compare the models and select the most appropriate one .

The best way to compare the various models is to find reliable existing machining data for the prediction of surface finish based upon the cutting parameters, in the form of tables, charts or graphs.

Holmes [1] developed a table for predicting surface finish values from given nose radius and feed rate values. General Electric Data Systems [2] gives a graph expressing surface finish as a function of nose radius for different feeds, as shown in Fig. 2.2 . Similarly the graph presented by GE Carboloy Systems [6] shows the same general trend but gives slightly different results, as shown in Fig. 2.3. The data presented in the graph by GE Carboloy systems is more recent, hence it is more consistent with the latest machining and measuring

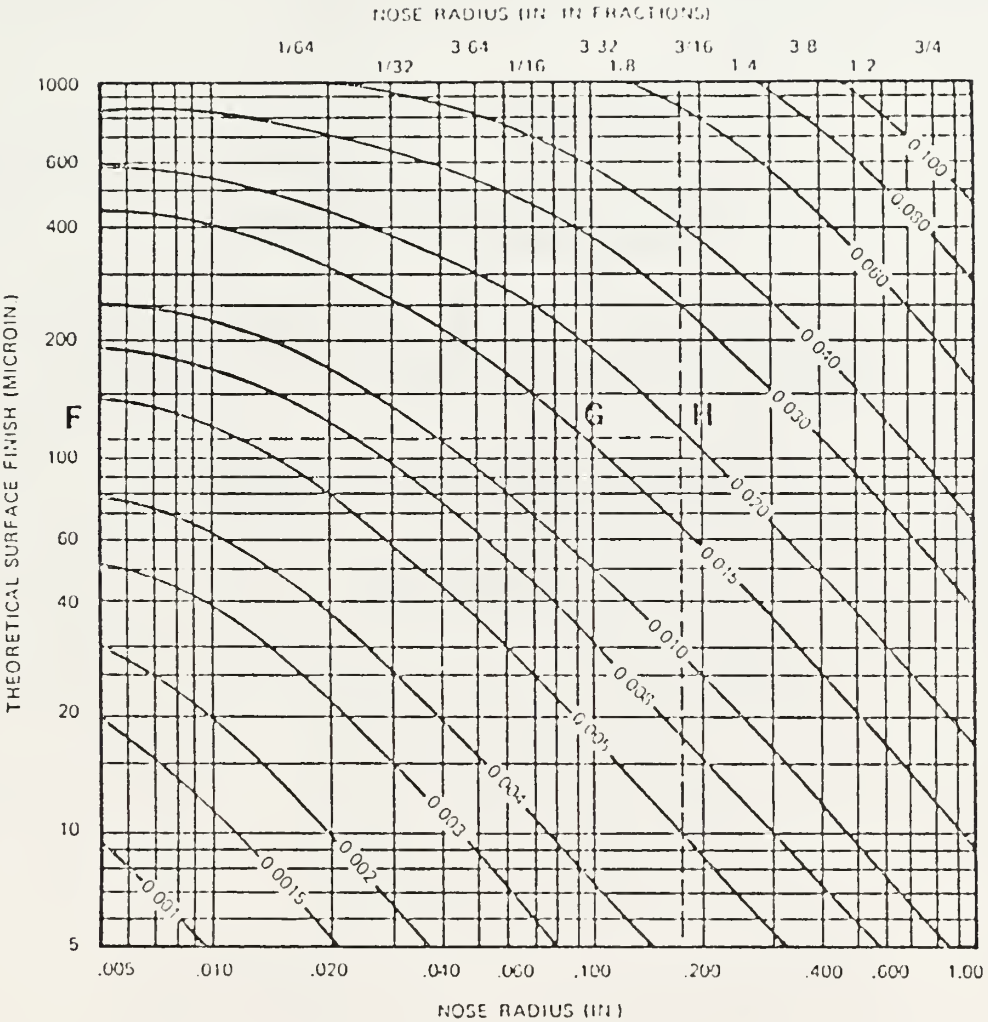



Figure 2.2 Relationship between surface finish, nose radius and feed. ( Reproduced from Ref. 2 )

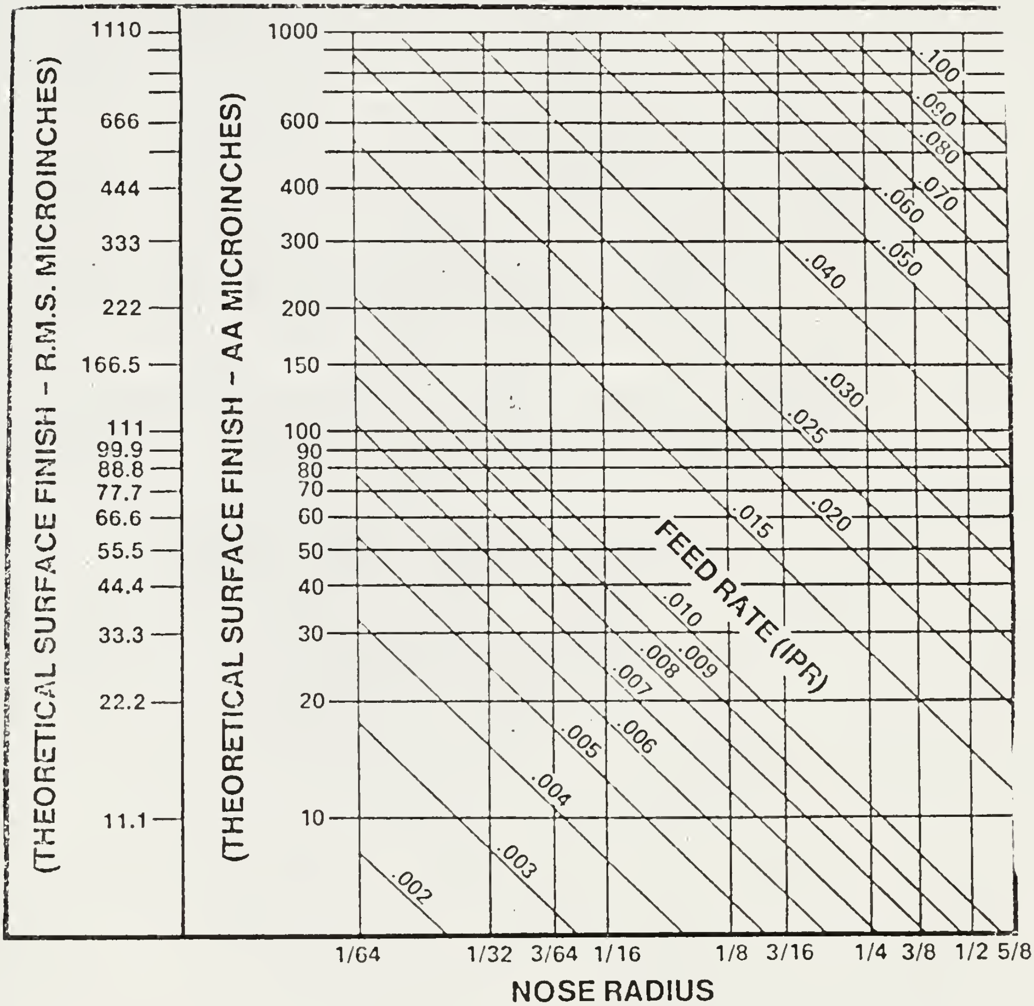



Figure 2.3 Theoretical Surface Finish as related to Nose radius and feed. ( Reproduced from Ref. 6 )

methods; it also provides a secondary graph that accounts for the effect of cutting speed on surface finish. Hence this machining data was chosen as the basis for the surface finish model.

Based upon the above discussions and as is evident from the graphs, the most important parameters affecting the surface finish are cutting feed rate and tool nose radius.

The nature of Fig. 2.2 and Fig. 2.3 and the other machining data make it clear that the surface finish must increase as feed increases and surface finish must decrease as nose radius decreases. Therefore the available machining data strongly favors the following functional dependency over that suggested by Eq. 2.4.

$$SF = K * F^2/RNOSE \tag{2.5}$$

where, SF = surface finish value predicted,

F = cutting feed rate ,

RNOSE = tool nose radius , and

K = a constant of proportionality.

A plot between surface finish and $F^2$/RNOSE (feed squared divided by nose radius) is drawn for various values of nose radius . This, as shown in Fig. 2.5, turns out to be a very narrow bunch of almost straight lines. This further confirms a linear relationship between the surface finish and $F^2$/RNOSE .

Thus it is concluded that the model for surface finish is of the form shown in Eq. 2.5. Great care must be taken in establishing a value for the constant K. The model must be conservative because it is

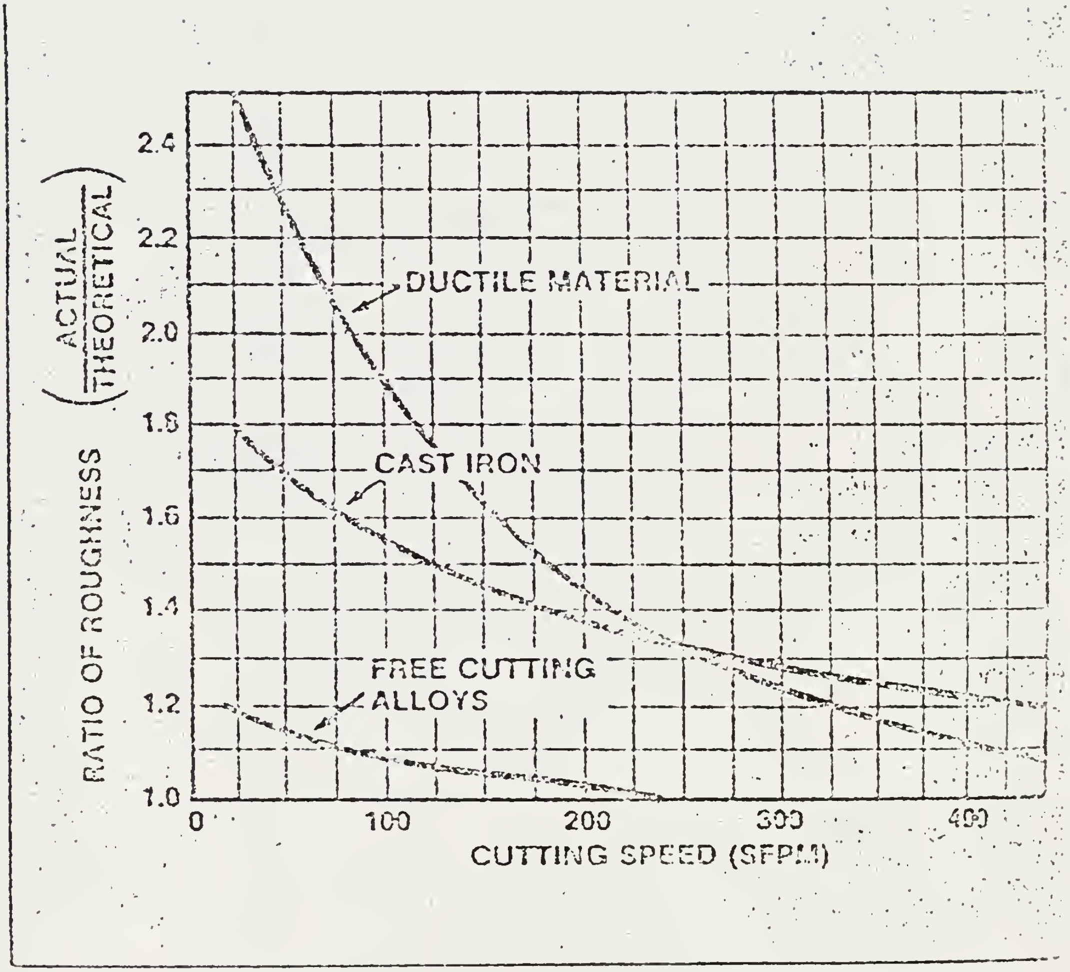



Figure 2.4 Effect of cutting speed on theoretical surface finish

(Reproduced from Ref. 6 )

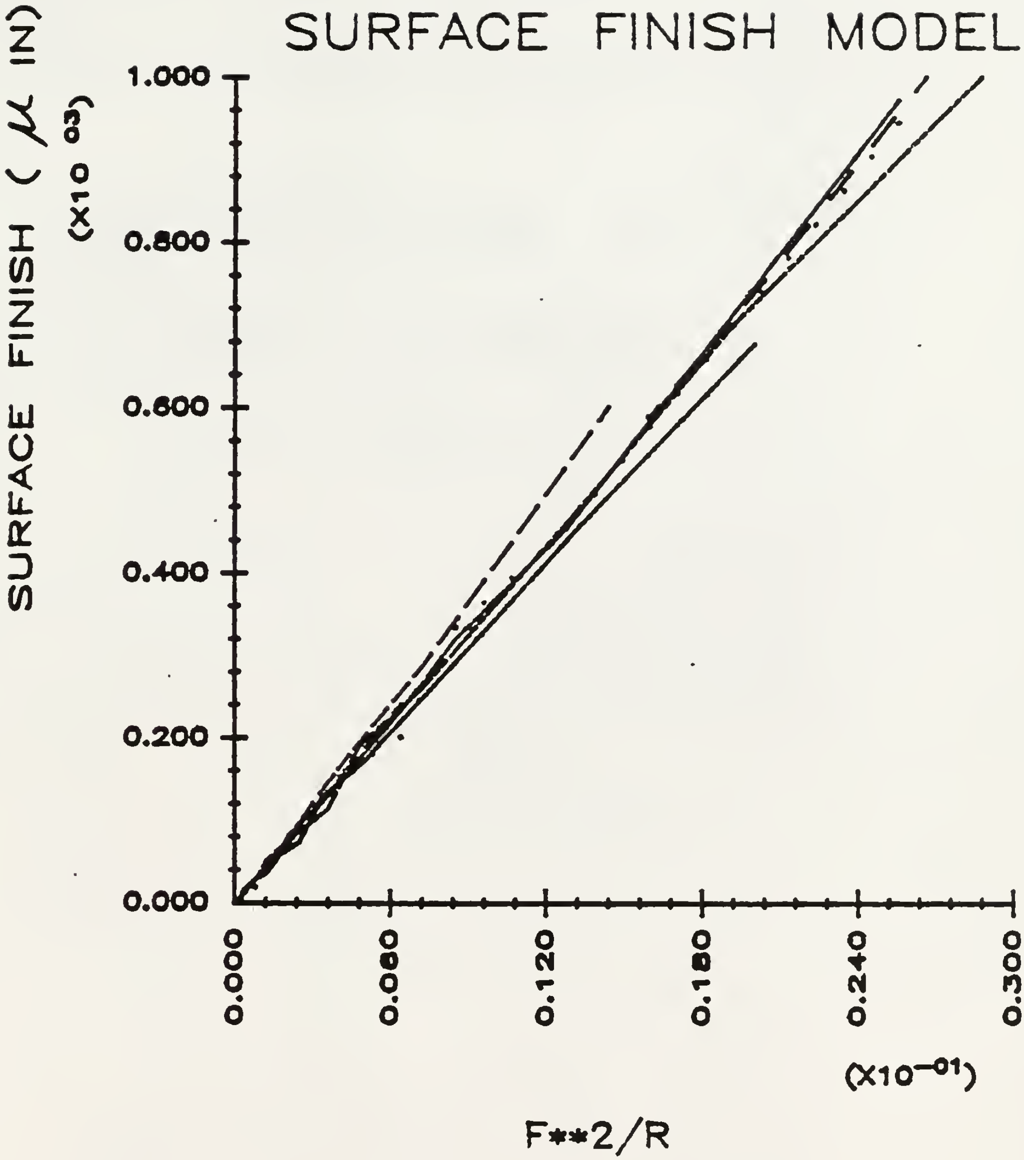



Figure 2.5 Development of the Surface Finish Model

misleading to predict finer surface finish values which may be in error, while it is acceptable to predict values which are slightly pessimistic; in the latter case, the model is acceptable because it ensures that the surface finish produced will be within specifications, even though the machining parameters may be set a little conservatively.

For the above reasons, the straight line at the extreme end in Fig. 2.5, predicting the coarsest surface finish is to be used in the model. The line is extended as shown and the slope is calculated. The value of the slope was found to be 4.16666 * 1. E04 . Thus,

$$SF = 4.16666 * 10^4 * F^2/RNOSE \qquad 2.6$$

where,

F = feed in in. per rev.

RNOSE = nose radius in in.

SF = predicted surface finish (RMS) micro in.

Now, to account for the effect of cutting speed, a speed correction factor has to be determined. Once again, the recent experimental data of [6] is considered. This reference gives graphs of the speed correction factor as a function of cutting speed for various types of materials such as ductile materials, cast iron and free cutting alloys. Generally ductile materials are used in the design of components; therefore a cubic polynomial was fit to the speed correction factor curve for ductile materials using the least squares method.The curve for the speed correction is reproduced in Fig. 2.4.

The resulting approximating function for the speed correction factor was :

$$Sp_f = A_1*(V^3) + A_2*(V^2) + A_3*(V) + A_4 \qquad 2.7$$

where ,

$Sp_f$ = Speed correction factor

$A_1$ = 2.755400325

$A_2$ = -0.010906436

$A_3$ = 0.000026424

$A_4$ = -0.000000023

and V is cutting speed in sfpm.

Thus the complete surface finish model can be written as

$$SF = Sp_f * K * F^2 /RNOSE \qquad 2.8$$

where, SF = predicted value of surface finish, micro in.

$Sp_f$ = speed correction factor

K = a constant given in Eq. 2.6,

F = Cutting feed rate of the part surface being turned, ipr.

RNOSE = Tool nose radius, in.

## 2.2 Tolerance Model :

The next important task is to develop a mathematical model for the prediction of manufacturing tolerance based upon the cutting parameters so that the tolerances on a component can be maintained within an acceptable range by suitable control of the machining parameters. Unfortunately past research in machinability has not satisfactorily addressed this issue which is of utmost concern, particularly in the manufacture of components with accurate mating requirements .

The size of a part from which all the dimensions are determined is the basic size. There are two extreme permissible sizes for a dimension. The largest permissible size for a dimension is called the upper or higher limit whereas the smallest size is known as the lower limit. The difference between the upper limit and the lower limit of a dimension is called the tolerance. When the tolerance is allowed on both sides of the nominal size, then the tolerance is said to be a bilateral tolerance whereas a unilateral system allows tolerances on one side of the nominal size only.

In the past there has not been much interest in including the manufacturing tolerances as independent design variables in the manufacturing design problems. Even though the tolerance of a part does not appear in the equation for manufacturing cost , the tolerances partly determine the cutting parameters, and therefore affect the manufacturing cost. Hence it becomes neccessary to seek relationships between part tolerances and cutting parameters.

After a search of existing machining data for predicting the manufacturing tolerance as a function of cutting parameters, only the data of Holmes [1] was found to be useful. This data is given in the form of a table relating minimum manufacturing tolerances to the feed factor and part diameter. This data is reproduced in Table 2.1.

VARIATION OF MINIMUM PART TOLERANCE WITH FEED FACTOR AND PART DIAMETER RANGE

| PART DIA. RANGE | FEED FACTOR 1.0 | .9 | .8 | .7 | .6 | .5 |
|---|---|---|---|---|---|---|
| upto .5 | .0010 | .0009 | .0008 | .0007 | .0006 | .0005 |
| .6 - 1.0 | .0012 | .0011 | .0010 | .0008 | .0007 | .0006 |
| 1.1 - 1.5 | .0014 | .00013 | .0011 | .0010 | .0009 | .0007 |
| 1.6 - 2.0 | .0016 | .0014 | .0013 | .0011 | .0010 | .0008 |
| 2.1 - 2.5 | .0018 | .0016 | .0015 | .0013 | .0010 | .0009 |
| 2.6 - 3.0 | .0020 | .0018 | .0016 | .0014 | .0012 | .0010 |
| 3.1 - 3.5 | .0022 | .0020 | .0018 | .0016 | .0013 | .0011 |
| 3.6 - 4.0 | .0024 | .0022 | .0019 | .0017 | .0014 | .0012 |
| 4.1 - 4.5 | .0026 | .0023 | .0021 | .0019 | .0016 | .0013 |
| 4.6 - 5.0 | .0028 | .0025 | .0022 | .0020 | .0017 | .0014 |
| 5.1 - 6.0 | .0032 | .0029 | .0026 | .0023 | .0019 | .0016 |
| 6.1 - 7.0 | .0036 | .0032 | .0029 | .0025 | .0022 | .0018 |
| 7.1 - 8.0 | .0040 | .0036 | .0032 | .0028 | .0024 | .0020 |

Table 2.1

(Reproduced from [1])

| | | | | | | |
|---|---|---|---|---|---|---|
| 8.1 - 9.0 | .0044 | .0040 | .0035 | .0031 | .0026 | .0022 |
| 9.1 -10.0 | .0048 | .0043 | .0038 | .0034 | .0029 | .0024 |
| 10.1-12.0 | .0052 | .0047 | .0042 | .0036 | .0031 | .0026 |
| 12.1-14.0 | .0056 | .0050 | .0045 | .0039 | .0034 | .0028 |
| 14.1-16.0 | .0060 | .0054 | .0048 | .0042 | .0036 | .0030 |
| 16.1-18.0 | .0064 | .0058 | .0051 | .0045 | .0048 | .0032 |
| 18.1-20.0 | .0068 | .0061 | .0054 | .0048 | .0041 | .0034 |

Diameter in inches,

Minimum Tolerance in inches.

Table 2.1 (CONTD.)

VARIATION OF MINIMUM PART TOLERANCE WITH FEED FACTOR AND PART DIAMETER

| PART | FEED FACTOR | | | | | |
|---|---|---|---|---|---|---|
| DIAMETER | 1.0 | .9 | .8 | .7 | .6 | .5 |
| 0.25 | .0010 | .0009 | .0008 | .0007 | .0006 | .0005 |
| 0.75 | .0012 | .0011 | .0010 | .0008 | .0007 | .0006 |
| 1.25 | .0014 | .00013 | .0011 | .0010 | .0009 | .0007 |
| 1.75 | .0016 | .0014 | .0013 | .0011 | .0010 | .0008 |
| 2.25 | .0018 | .0016 | .0015 | .0013 | .0010 | .0009 |
| 2.75 | .0020 | .0018 | .0016 | .0014 | .0012 | .0010 |
| 3.25 | .0022 | .0020 | .0018 | .0016 | .0013 | .0011 |
| 3.75 | .0024 | .0022 | .0019 | .0017 | .0014 | .0012 |
| 4.25 | .0026 | .0023 | .0021 | .0019 | .0016 | .0013 |
| 4.75 | .0028 | .0025 | .0022 | .0020 | .0017 | .0014 |
| 5.50 | .0032 | .0029 | .0026 | .0023 | .0019 | .0016 |
| 6.50 | .0036 | .0032 | .0029 | .0025 | .0022 | .0018 |
| 7.5 | .0040 | .0036 | .0032 | .0028 | .0024 | .0020 |
| 8.5 | .0044 | .0040 | .0035 | .0031 | .0026 | .0022 |

Table 2.2

| | | | | | | |
|---|---|---|---|---|---|---|
| 9.5 | .0048 | .0043 | .0038 | .0034 | .0029 | .0024 |
| 11.0 | .0052 | .0047 | .0042 | .0036 | .0031 | .0026 |
| 13.0 | .0056 | .0050 | .0045 | .0039 | .0034 | .0028 |
| 15.0 | .0060 | .0054 | .0048 | .0042 | .0036 | .0030 |
| 17.0 | .0064 | .0058 | .0051 | .0045 | .0048 | .0032 |
| 19.0 | .0068 | .0061 | .0054 | .0048 | .0041 | .0034 |

Diameter in inches,

Minimum Part Tolerance in inches

Table 2.2 (CONTD.)

Feed factor is defined as the ratio of the actual cutting feed rate to the nominal feed rate for the part material.

$$Feed_f = (feed_{act}/feed_{nominal}) \qquad 2.9$$

$feed_{act}$ is the actual cutting feed rate of the surface being machined

$feed_{nominal}$ is the maximum possible feed rate for the tool and job materials being used and is selected based upon the past experience of the user or machinability data such as Holmes[1].

For the tolerance model to be continuous, it should be able to predict minimum tolerances at all diameter values. Hence the diameter ranges were replaced by mean diameter values by substituting the average of the range as the representative diameter for each plot. Now the lines can be considered to represent the minimum tolerance values based on feed factor, for various continuous diameter values. The modified model is shown in Fig. 2.6 and 2.7 . Since the plots are linear, the function for each curve should be of the form

$$TOL = m_1 * Feed_f + c_1 \qquad 2.10$$

where, TOL is the predicted value of the minimum part tolerance,

$m_1$ is the slope which depends on the part diameter , and

$c_1$ is a constant taken to be zero based upon the fact that for a feed of zero value , a non zero tolerance value is meaningless ; further the trend of the plots in Fig. 2.6 and

Fig. 2.7 justifies the zero value of $c_1$.

After it is known from the plots in Fig.2.6 and 2.7 that the slope $m_1$ of Eq. 2.10 depends on the value of part diameter , the next step is to find this relationship . For this purpose a plot is drawn between the slope values $m_1$ and the corresponding part diameters, as shown in Fig.2.8 . This plot, as is clear from the figure, is in the form of a broken straight line which can be represented as :

$$m_1 = m_2 * DIA + c_2 \qquad 2.11$$

with $m_2$ and $c_2$ being represented as follows :

$m_2 = .4$ for $2.5 \leq DIA \leq 9.5$ in

$c_2 = 0.9$ for $2.5 \leq DIA \leq 4.5$ in

$c_2 = 1.0$ for $5.5 \leq DIA \leq 9.5$ in

where DIA is part diameter in inches.

Now, in order to approximate this curve by a single linear function the more conservative of the values $c_1$ , $c_2$ should be considered for the whole range of diameter values. Thus, we should use Eq.2.11 with $m_2$ set to 0.4 and $c_2$ set to 1.0 to obtain $m_1$. Substituting this expression for $m_1$ into Eq. 2.10 we obtain the following equation for the prediction of minimum tolerance :

$$TOL = [\ 1.0 + .4*DIA] * 1.0\ E\text{-}03 * feed_f \qquad 2.13$$

where, TOL is the predicted value of minimum tolerance ,

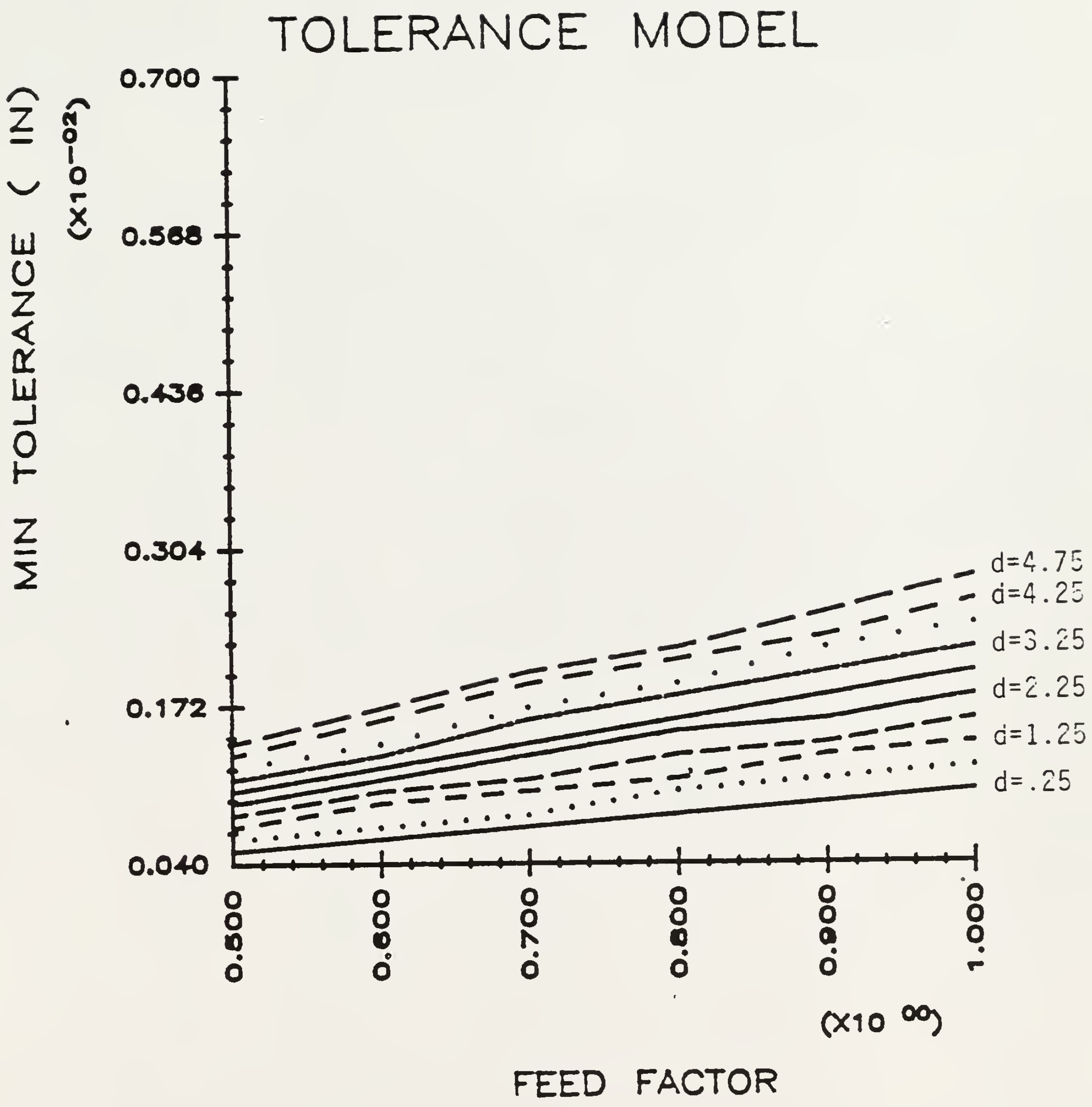



Figure 2.6 Minimum Part Tolerance as a function of feed factor for various values of part diameter from 0.25-4.75 in.

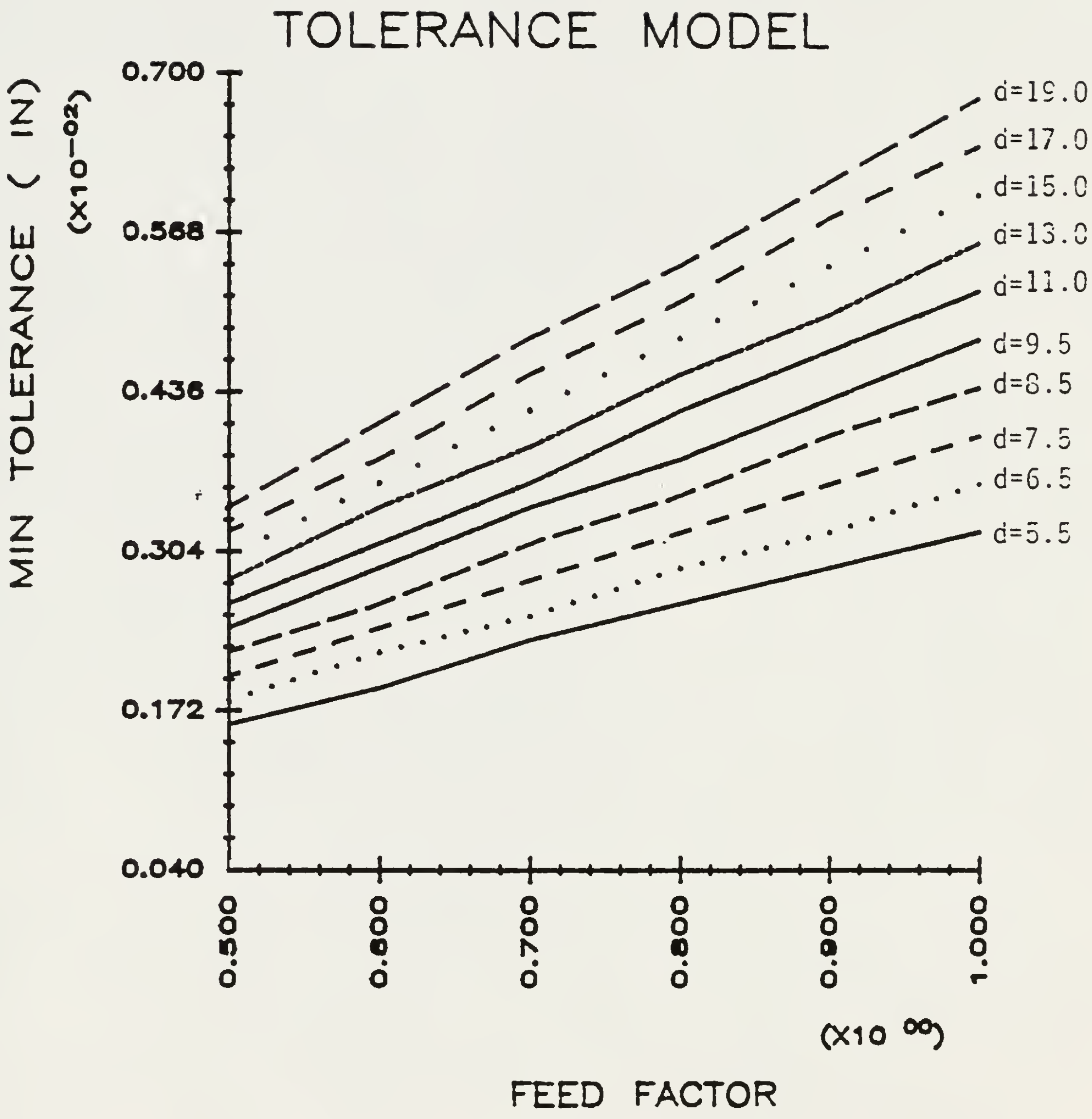



Figure 2.7 Minimum Part Tolerance as a function of feed factor for various values of part diameter from 5.5 to 19.0in

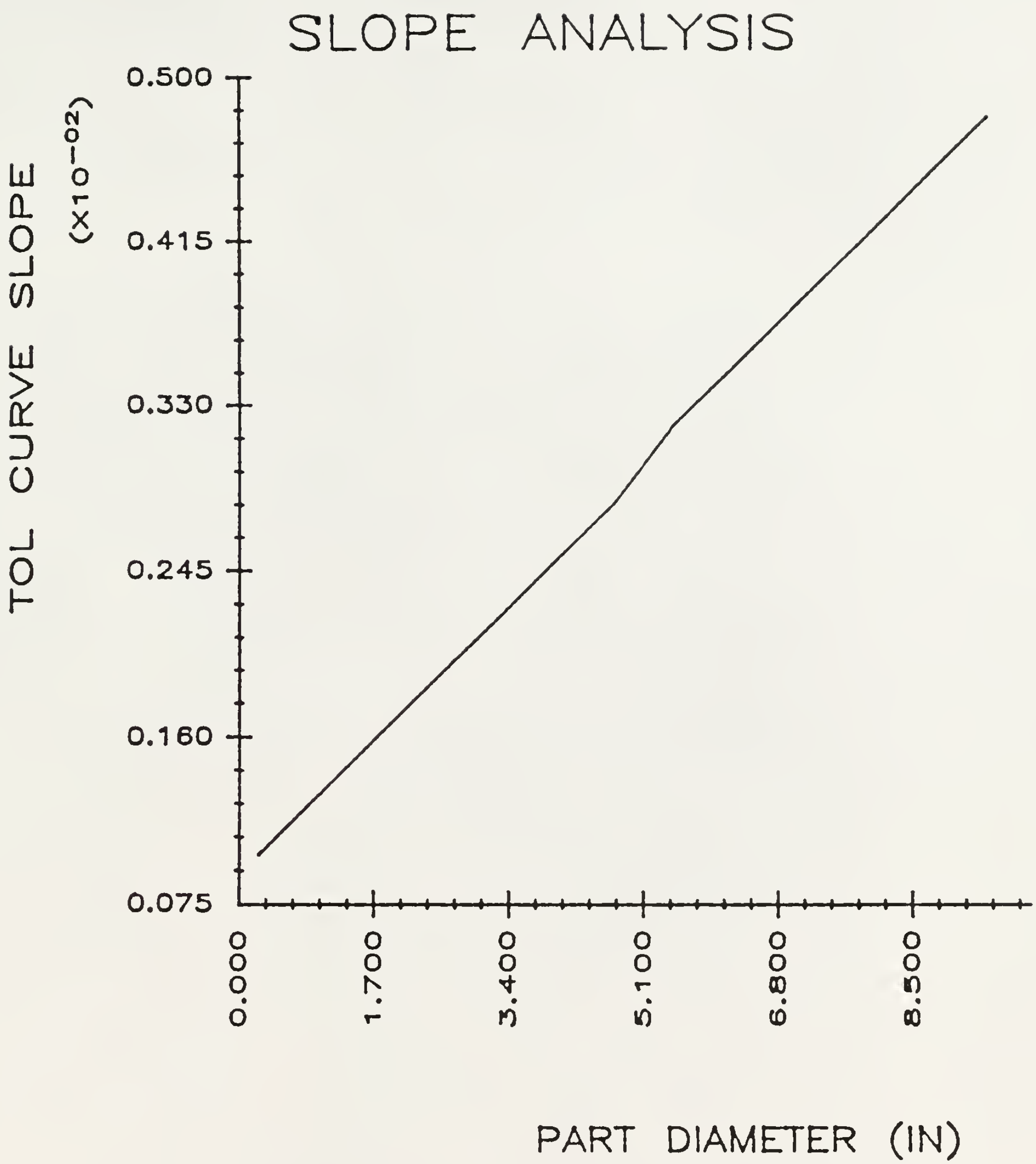



Figure 2.8 Relationship between the slope of the tolerance model and part diameter values.

DIA is the diameter of the part,

$feed_f$ is the feed factor given by Eq. 2.9

The maximum possible feed rate for various materials can be selected either by the user based upon past experience or from machinability data [1].

The tolerance model derived above is a reasonably accurate fit of existing tables and graphs. Further, it is a continuous model suitable for usage in mathematical programming formulations.

## 2.3 Tool Life Equation :

The machining time is known to decrease with increased speed and feed ; however the tool wear increases as well and so tool life shortens rapidly. It is thus evident that the tool life is an important factor in any manufacturing model.

Taylor [7] ran extensive tests to determine the relationship between cutting speed and tool life for turning operations and found that when cutting speed is plotted against tool life on log-log axes a straight line results in the region of normal cutting speeds, i.e. we may expect a relationship of the form :

$$\log V = - n * \log T + \log C \qquad 2.14$$

where, V is the cutting speed for turning of the part in sfpm ,

T is the tool life in minutes,

n is the slope of the straight line plot,

C is a constant for a given combination of cutting conditions, expressing the speed for a tool life of 1 minute.

TYPICAL TOOL-LIFE CONSTANTS

| Work Material | coefficient C | coefficient n |
| --- | --- | --- |
| Brass(60 Cu-40 Zn) | 299 | .096 |
| Bronze(90 Cu-10Sn) | 232 | .111 |
| SAE-1112 | 225 | .105 |
| SAE-2340 | 143 | .147 |
| SAE-3140 | 299 | .096 |
| SAE-4140 | 232 | .111 |
| Cast Iron (160 Bhn) | 225 | .105 |
| Cast Iron (205 Bhn) | 143 | .147 |
| Monel Metal | 299 | .096 |
| SAE-3240 (annealed) | 232 | .111 |
| Cast Iron (200 Bhn) | 225 | .105 |
| SAE-1060 (annealed) | 143 | .147 |
| SAE-2340 (annealed) | 299 | .096 |
| SAE-4147 H (230 Bhn) | 232 | .111 |
| AISI-81B45 | 225 | .105 |
| Cast Iron | 143 | .147 |

Table 2.3 Typical Tool-life constants

(Reproduced from [7])

Values of n and C for some cutting situations as given by J. P. Vidosic [7] are reproduced in Table 2.3 .

Now, Eq. 2.14 can be written as

$$V = T^{-n} * C \qquad 2.15$$

which can be rearranged to yield

$$V\ T^{n} = C \qquad 2.16$$

where the parameters are same as in Eq. 2.12.

Taylor's equation for tool life is a popular one and is the basis of most of the important data and related calculations on tool life. Other researchers have attempted to develop modified tool life equations based on Taylor's equation. Gilbert and Truckenmiller [8] advocate the use of the relationship

$$V\ T^{0.125} = K_{tm}\ K_{mc}\ /f^{0.61}\ d^{0.36} \qquad 2.17$$

where, $K_{tm}$ = constant for tool life depending upon tool material

$K_{mc}$ = constant for tool life, depending upon material cut

f = feed (inches per revolution)

d = depth of cut (in.)

T = tool life (min.)

V = speed in feet/min

Hati and Rao [5] used the following equation :

$$T = (\alpha\ V^{\alpha 1}\ f^{\alpha 2}\ \ d^{\alpha 3}\ ) \qquad 2.18$$

where, T is the tool life, min.

V is the cutting speed, m/min

f is the feed, mm/rev

d is the depth of cut, mm.

$\alpha$, $\alpha1$, $\alpha2$, $\alpha3$ are constants depending upon the tool piece combination

Although these equations attempt to include the effects of feed and depth of cut while estimating tool life and are more elaborate, the present work only makes use of the standard Taylor's equation for a variety of reasons. These reasons include the ready availability of related coefficients and the fact that the depth of cut is not a critical factor in our case since we are primariliy interested in the finishing cut.

Thus the basic mathematical models for the prediction of surface finish and manufacturing tolerance for the part surfaces being machined have been developed. The tool life equation for the turning operation has also been selected. These models will be extensively used in later chapters for the determination of manufacturing cost and the application of constraints to the optimization problem stated in Eq. 1.1 .

# CHAPTER III

## OPTIMIZATION OF THE MANUFACTURING PROCESS

A large number of engineering problems have been solved successfully by the application of optimization techniques. These techniques, after a series of iterative numerical calculations, provide the user with design modifications which must otherwise be based on the designer's intuition and experience. In order to determine optimum machining parameters for the minimization of manufacturing cost, the problem has to be reduced to a standard nonlinear programming problem of the form expressed in Eq. (1.1) , which is repeated here for convenience.

$$\begin{aligned} &\text{Minimize} &:\quad & F(B) \\ &\text{Subject to} &:\quad & g_j(B) \le 0.0 \quad ; j=1,m \quad \text{(inequality constraints)} \\ & & & g_l(B) = 0.0 \quad ; l=m+1,n \quad \text{(equality constraints)} \end{aligned} \tag{3.1}$$

where B is the design vector containing the design variables $[b_1, b_2, b_3, \ldots\ldots\ b_{nv}]^T$ with nv being the number of design variables,

n is the total number of constraints.

In this chapter the process of converting the manufacturing problem to an optimization problem is discussed. This process involves (i) selection of design variables, (ii) the formulation of the cost function, and (iii) the formulation of the constraint functions.

## 3.2 Design Variables :

Based upon past experience, it can be stated that the important design variables in this class of manufacturing problems are the cutting feed rate and the cutting speed. These variables are under the control of the machinist; the machinist tries to obtain desired characteristics of the manufactured components by selecting appropriate values for these variables. However, some other parameters in these manufacturing design problems, though important, are not directly under the control of the machinist. These parameters include part tolerances, surface finish, tool life, etc. After a thorough consideration of these parameters, the design vector for the current optimization problem of minimizing manufacturing cost was chosen to include the following as design variables :

(i) cutting speeds ,

(ii) cutting feeds , and

(iii) manufacturing tolerances .

The present work considers the minimization of manufacturing cost of machined components. The components are assumed to be roughly turned to a reasonable size from the raw stock. Such components have to be turned to the required final dimension accurately by a finishing cut. Hence the depth of cut is not critical and does not appear in the list of design variables.

3.3 Cost Function :

In order to manufacture goods at either minimum cost or maximum production rate, it is necessary to control all the essential parameters of the manufacturing process. An important step in this task is to arrive at a reasonable cost function. The total cost is the sum of the loading/idling cost, the cutting cost, the tool changing cost and the tool regrinding cost. For turned components [2] each of these costs is determined as follows :

$$\text{Idle cost/piece} = K_1 * \text{Idle time/piece} \qquad 3.2$$

$$\text{Cutting cost/piece} = K_1 * \text{Cutting time/piece}$$

$$= K_1 *(L \pi D)/(12fV) \qquad 3.3$$

$$\text{Tool change cost/piece} = K_1 *(\text{Tool failures/piece}) * TCT$$

$$= K_1 *(L \pi D V^{1/n-1})(TCT)/(12f C) \qquad 3.4$$

$$\text{Tool regrinding cost} = K_2 *(\text{tool failures/piece})$$

$$= K_2 *(L \pi D V^{1/n-1})/(12 f C) \qquad 3.5$$

where,

$K_1$ = direct labor rate plus overhead rate in $/min, including operator and helper labor, maintainance, power, depreciation, and insurance;

$K_2$ = Tool cost per grinding, including original and regrinding costs in dollars per tool

L = Length of part in inches

D = Diameter of part in inches

V = Cutting Speed in sfpm

f = Feed in inches per revolution

C = Cutting speed for one minute tool life

TCT = Tool-change time in minutes

The total cost will be equal to the sum of the individual costs of all the parts being manufactured for any machine assembly. For example if the shaft assembly shown in Fig. 3.1 is to be produced, then the final cost function equals the sum of the cost functions for (i) the shaft, (ii) the two bearings, and (iii) the housing. The manufacture of shaft assemblies of this type is chosen as the class of problems on which the proposed optimization techniques will be tested.

The cost function for the machining of a single shaft can be expressed as the sum of the costs in equations 3.2, 3.3, 3.4 and 3.5 for the case of the shaft, as indicated below :

$$f = K_1[\text{ idle time }] + K_1[L \pi D]/(12f\ V) + K_1[L \pi D/(12f)] *(V/C)^{1/n}(1/V)(TCT) + K_2[ L \pi D /(12f)]*(V/C)^{1/n}(1/V) \quad 3.6$$

Since the idle time does not have any cutting parameter or design parameters involved, the constant contribution of the first term can be neglected. Thus, the modified expression for the cost function becomes :

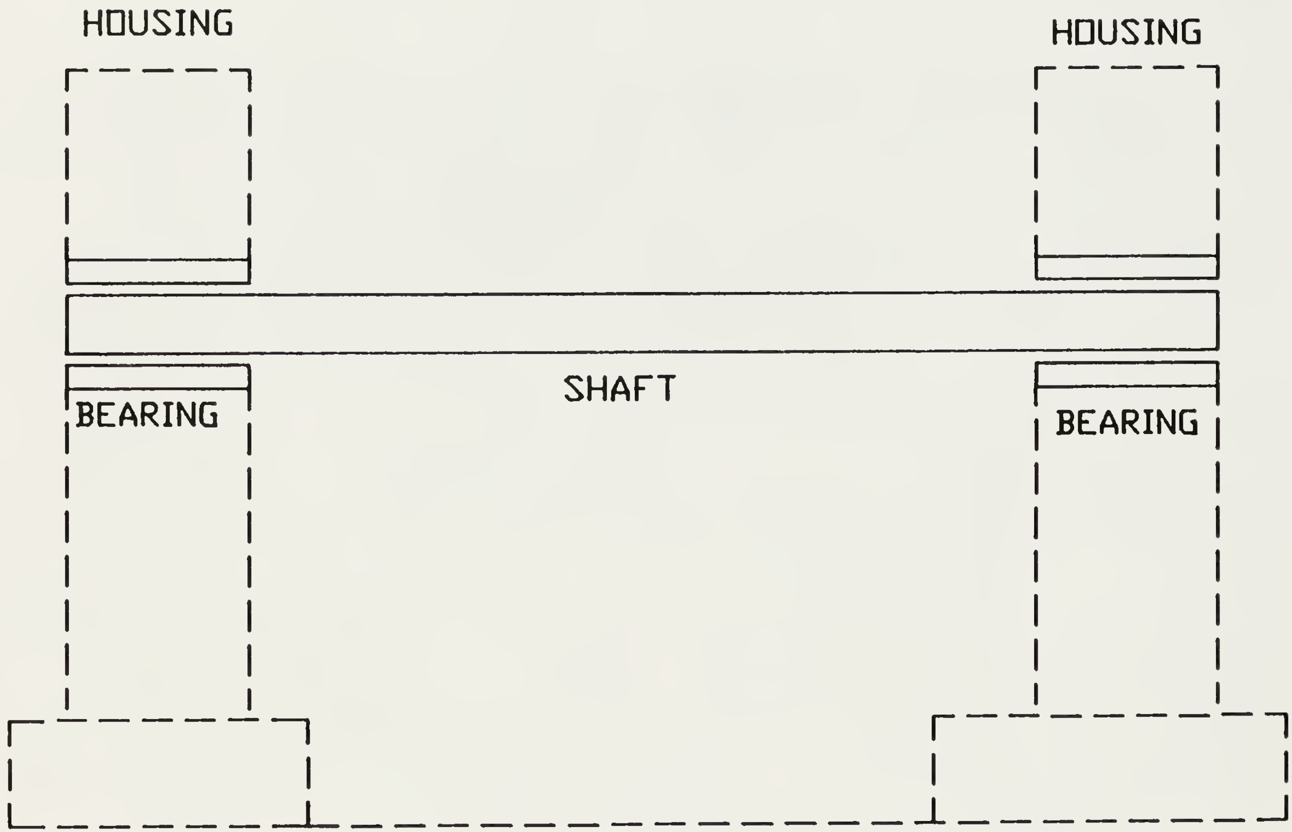



Figure 3.1

$$f = K_1 \ [(L \pi D)/(12fV)](\ 1 + (V/C)^{1/n}(TCT + K_2/K_1\ )\ ] \qquad 3.7$$

Thus for the shaft alone, we have

$$f_{shaft} = [\ \pi/12 * LD/fV * (K_1 + (K_1*TCT+K_2)*(V/C)^{1/n}) \qquad 3.8$$

Similar terms for other related components must be added to get the total cost function. Thus, for the assembly shown in Fig. 3.1 , we obtain ,

$$F = f_{shaft} + 2\ f_{bearing} + 2\ f_{housing} \qquad 3.9$$

If there are more shafts in the assembly, then the total cost becomes

$$FF = \sum F_i \ , \qquad \text{where } i = 1, \text{ number of shafts} \qquad 3.10$$

In the numerical examples, manufacture of shaft assemblies of this type are considered in detail. These shaft assemblies are commonly used for the transmission of power at various torques and speeds. Each shaft is supported in bearings at the ends and power loads are encountered on the shafts due to the presence of pulleys and gears.

## 3.4 Formulation of Constraint Functions

The problem of determining the optimum cutting parameters has been tackled by several researchers. Unfortunately, most efforts in this area do not treat manufacturing constraints satisfactorily. Ermer[16] solved the constrained machining economics problem by using geometric programming, but did not include the constraints necessary to ensure tolerance and surface finish requirements. Hati and Rao [5] also applied constraints to the machining economics problem but most of the constraints were simply bounds on parameters like feed, speed and depth of cut. Bhattacharya [4] presented the first successful application of constraints to guarantee satisfaction of surface finish requirements. This model also included constraints to impose bounds on speeds and feeds. However constraints relating to tolerances were not included.

In the present work, a very large number of constraints can be imposed to force the design vector of machining parameters to an optimal value which satisfies all surface conditions and tolerance requirements. The tolerance requirements may include conditions on fits between mating parts in addition to limits on individual tolerances. The standard set of constraints used in the optimization of the manufacturing process are the following :

First of all, the bounds on all design variables are user input values constraining the range of design variables. These are referred to as the bound constraints on the design variables.

The next set of constraints consists of those that impose tolerance requirements on each diameter. The conditions that must be met here are that the minimum tolerance predicted by the model in chapter II on each diameter must be within the tolerance specified on that diameter. Since the tolerance model gives a conservative prediction, this will ensure that the cutting parameters chosen will satisfy the required tolerances.

There are two ways of expressing the tolerance on a part diameter. The first one is to give the nominal diameter and the part tolerance ; the actual part dimensions in this case may vary as expressed below.

$$\text{Maximum limit} = \text{Nominal diameter} + \text{Tolerance}$$

$$\text{Minimum limit} = \text{Nominal diameter} - \text{Tolerance} \qquad 3.11$$

In the second approach , the upper tolerance and the lower tolerance are explicitly stated along with the basic size. The actual part dimension in this case may vary as follows :

$$\text{Maximum limit} = \text{Basic size} + \text{upper tolerance}$$

$$\text{Minimum limit} = \text{Basic size} + \text{lower tolerance} \qquad 3.12$$

The minimum tolerance prediction model was developed using the first approach as the data available for the prediction of manufacturing tolerance [1] uses this method. Also the mathematical modeling of the tolerance prediction is convenient by this method as only one tolerance value is to be stated besides the nominal diameter. In the second approach for every basic diameter, both upper and lower tolerance values are to be given. At the same time , from the manufacturer's point of view it is more precise to mention the upper and lower tolerance values, hence manufacturing specifications follow the second approach. For the same reasons the manufacturing tolerance constraints should be expected to be in the second form and hence the constraint functions use this approach. Fortunately the two ways of expressing the tolerance can be made compatible by setting the tolerance value in the first approach to be equal to the mean of the upper and lower tolerances in the second approach. This comparison becomes necessary when using both types of tolerance specifications in the same constraint equation as in the case of the tolerance constraints described in the next section.

## 3.4 Formulation of Tolerance Constraints

The tolerance constraint is of the following form :

$$TOL \leq \text{Part Tol}_{act} \qquad 3.13$$

where TOL is the value of minimum part tolerance predicted by the tolerance model developed in the previous chapter, based upon the machining parameters being used in the turning of the part,

Part $\text{Tol}_{act}$ is the actual part tolerance depending upon the values of the tolerance design variables; it is expressed as

$$\text{Part Tol}_{act} = (UTOL-LTOL)/2 \qquad 3.14$$

where UTOL,LTOL represent the upper and lower tolerances of the part diameter.

Thus the tolerance constraint can now be stated in standard form as :

$$TOL - \text{Part Tol}_{act} \leq 0. \qquad 3.15$$

Using the tolerance model developed in the previous chapter, the above constraint equation for the outer surface of the shaft becomes

$$(1.0 + 0.4(DIAOS))(FDOS/(1000.(NFDOS)) - (ABS(UTOLOS - LTOLOS))/2. \leq 0.0 \qquad 3.16$$

where, UTOLOS represents the upper tolerance on shaft outer surface,

LTOLOS represents the lower tolerance on shaft outer surface,

NFDOS is the maximum recommended cutting feed for the shaft material as explained in the tolerance model in chapter II

DIAOS is the outside diameter of the shaft,

FDOS is the cutting feed for turning the shaft outer surface.

Similar equations exist for the inner surface of the bearing, the outer surface of the bearing and the surface of the housing hole for each shaft. The two bearings on any particular shaft are assumed to have the same dimensions and need not be considered individually. The same is true of any two holes in the housing that correspond to the same shaft.

The third set of constraints are those forcing the tolerance design variables to be distributed in such a way so as to satisfy the fit requirements between mating surfaces. The relationship between fits and tolerances must be fully understood in order to formulate the constraints, hence a discussion is presented below.

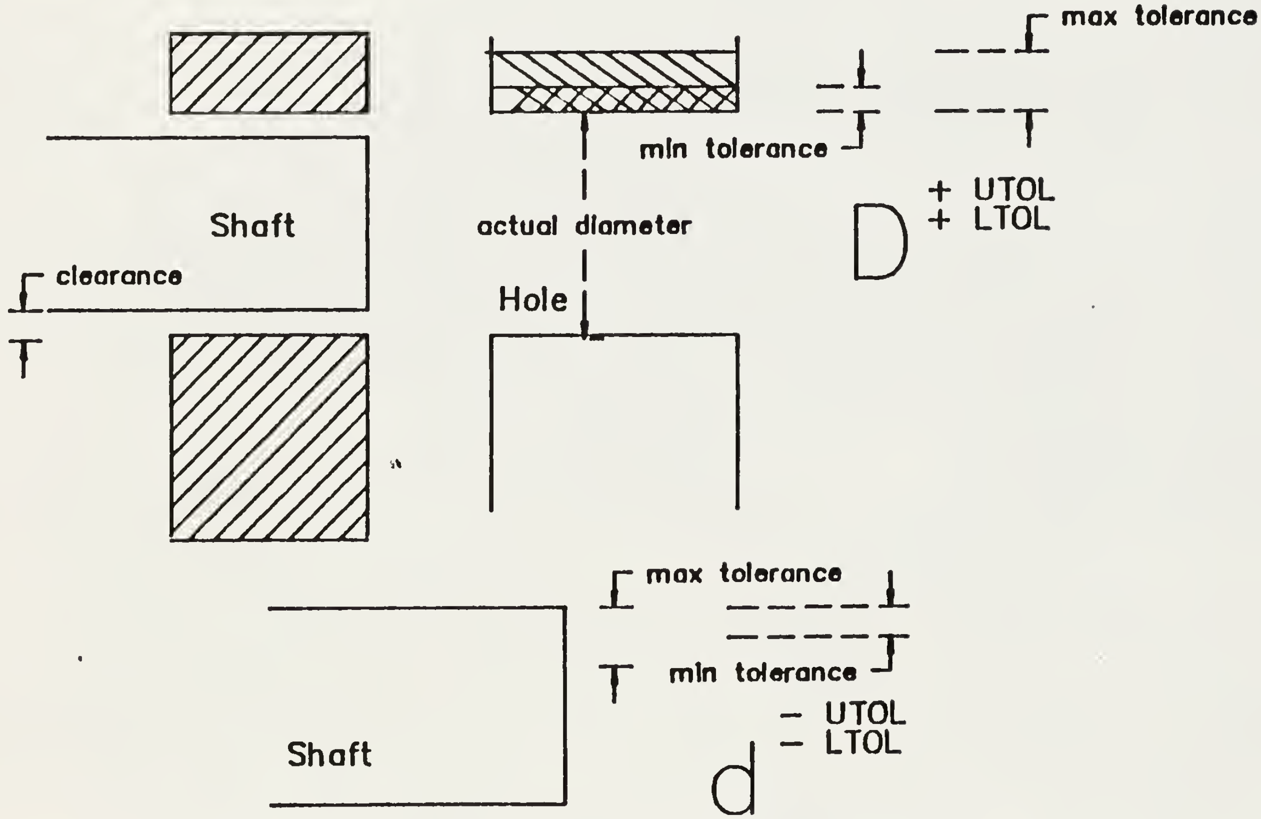
Tolerance specifications are considered to be either shaft based or hole based. In the hole based system, tolerances are positive values and are added to the basic diameter of the hole to give the upper and lower limits.

$$TOLN = (UTOL-LTOL)/2 \qquad 3.17$$

where, TOLN is the part diameter tolerance,

UTOL is the upper tolerance value on the part diameter, and

LTOL is the lower tolerance value .




HOLE & SHAFT SYSTEM OF TOLERANCES

Figure 3.2

In the case of the shaft based systems, tolerance values are considered to be negative values and thus are substracted from the basic size to get the limits.

TOLN = ( ABS(LTOL)-ABS(UTOL) )/2 3.18

Fig. 3.2 shows the the hole and shaft systems.

Based upon the fit between the mating parts there are three kinds of fits.

1. Clearance fits :- For these types of fits the mating parts are so toleranced that clearance between them always occurs. Examples of clearance fits are slide fits, easy sliding fits, running fits, slack running fits and loose running fits.

2. Interference fits :- In interference fits the mating parts are so toleranced that interference between them always occurs. Examples are shrink fits, heavy drive fits and light drive fits.

3. Transition fits :- In this type of fit, the selection of tolerances on mating parts is such that either clearance or interference may occur depending upon the actual size of the mating parts. Transition fits are used for force fits, tight fits and push fits.

The fit between two mating parts is described by the maximum clearance and the minimum clearance. Also, the maximum possible gap between the hole and the shaft will be the sum of the upper limit of the hole and the lower limit of the shaft . In other words the combination of the biggest hole and the smallest shaft diameter is the case in which the maximum possible gap is observed. If this maximum gap is guaranteed to be less than the maximum clearance specification of the fit then the tolerance distribution can be said to satisfy the maximum clearance constraint. Similarly, the constraint to force the least possible gap to be greater than the minimum clearance of the fit can also be written in terms of the tolerances. The case in which the hole is the smallest possible and the shaft diameter is the largest is the case of minimum possible gap. These constraints can be written as :

$$(UTOLIB + | LTOLOS |) - MXCSB \leq 0 \qquad 3.19$$

where MXCSB is the maximum clearance between shaft and bearing.

$$MNCSB - (LTOLIB + | UTOLOS |) \leq 0 \qquad 3.20$$

where MNCSB is the minimum clearance between shaft and bearing.

Similar sets of constraints are applied for fits between the bearing outer surface and the inner surface of the housing.

In addition to these constraints, further constraints are needed to maintain minimum separation between an upper tolerance and the corresponding lower tolerance. This separation is specified by the user, based on estimates of the most demanding tolerances that can be permitted.

The constraint functions that impose these conditions on the shaft outer surface are of the form :

$$ABS(UTOLOS-LTOLOS)/2 - TOLOS \leq 0 \qquad 3.21$$

where, TOLOS is the desired tolerance separation and

UTOLOS, LTOLOS are the design variables which are directed to the feasible range through this constraint.

Similar equations are used for all other surfaces being machined like the inside surface of the bearing, the outside surface of the bearing and the inside surface of the housing.

The final set of constraints ensures the required surface finish. These constraints make use of the surface finish model described in chapter II and are of the form :

$$SF_{pred} - SF_{des} \leq 0 \qquad 3.22$$

where, $SF_{pred}$ is the value of the surface finish predicted by the cutting parameters through the surface finish model of chapter II ,

$SF_{des}$ is the value of the surface finish as desired by the user

Substitution of the predicted surface finish from the model results in :

$$\frac{K(FDOS)^2}{RMXOS} - SF_{des} \leq 0 \quad \text{(for shaft surface)} \qquad 3.23$$

where, RMXOS is the maximum allowable tool nose radius for turning the shaft,

K is the constant described in chapter 2,

FDOS is the value of the feed for turning the shaft outer surface.

The predicted surface finish is the maximum possible roughness of the surface being turned with the given machining parameters. Since the nose radius of the tool does not occur in the cost function or in any other constraint functions, it does not have to be treated as a design variable. Instead, the user is required to specify the maximum allowable value (upper bound) for the nose radius and this is used in the surface finish constraint of Eq. 3.22 . After the optimization has been completed and all the constraints have been satisfied, the actual nose radius to be used can be calculated by rewriting Eq. 3.22 as

$$NROS = K[FDOS^2]/SF_{des} \qquad 3.24$$

where, NROS is the value of the optimum nose radius of tool,

K, FDOS , $SF_{des}$ are as defined earlier.

Since the constraint was satisfied using the largest allowable nose radius, it follows that the nose radius calculated by this formula will be smaller than the maximum allowable value, and will satisfy the surface finish constraint exactly.

The preceding discussion makes it clear that a large number of constraints must be imposed to ensure that the cutting parameters chosen satisfy manufacturing requirements like surface finish, mating requirements, tolerance requirements, etc. Without these constraints being satisfied the optimization is not meaningful and may lead to the selection of a set of cutting parameters which will not be acceptable. In cases where the user does not wish to impose certain constraints, the input values can be set such that the influence of those constraints is greatly reduced. For example if a very coarse surface finish value is specified for a component surface, the corresponding constraint can be very easily satisfied and so it will not affect the optimization beyond a few iterations.

# CHAPTER IV

# COMPUTER INTEGRATED OPTIMIZATION OF DESIGN AND MANUFACTURE

## 4.1 Introduction :

Computers are now extensively used for problem solving in all disciplines of engineering because of their enormous capacity to store and manipulate large volumes of data in a very efficient manner. Manufacturing and design are among the fields of engineering that have been radically transformed by the advent of these machines. In fact, the emerging field of Computer Integrated Manufacture (CIM) is a direct result of the computerization of design and manufacture. CIM links the design and manufacturing aspects of engineering into a single unit. Engineers working in this area have now reached a stage where machining data and design considerations are stored in computers which can directly control manufacturing facilities like NC machine tools and robots to machine a part to high accuracy in relatively low manufacturing time. In addition, these machines have the capacity to repeat the process any number of times. However, CIM is currently at a stage where it lacks the ability to take manufacturing considerations into account at the design stage itself. Also, the set of cutting parameters that is used is seldom optimized.

In this chapter, a method for simultaneous optimization of design and manufacture is presented.

As already stated in chapter I the design of the component has a direct effect on the manufacturing cost and the design of a component should be influenced at least partially by manufacturing considerations. The diameters of shafts, the distribution of loads (gears/pulleys) on a shaft, the dimensions of bearings etc. are to be selected in such a way that the manufacturing cost is minimized. It was mentioned in chapter I that the distribution of the loads should not be fixed arbitrarily by hit and trial methods ; rather, mathematical constraints should be applied to enforce all design requirements and these constraints should be incorporated in the optimization problem for cost minimization. This is achieved by considering (i) the machining parameters, and (ii) the component design parameters as design variables for the optimization problem and applying both manufacturing and design constraints simultaneously.

For shaft assembly of the type shown in Fig. 1.1, the set of variables that describe the design of the system are the following :

(i)outer diameter of the shaft ,

(ii)outer diameter of the bearing ,

(iii)bearing length ,

(iv)the load distribution (distances of horizontal, vertical loads, i.e. the distance of each load from a fixed end of the shaft).

For design calculations, it is clear from Fig. 1.1 that the inside diameter of the bearing can be considered equal to the outside diameter of the shaft; similarly the inside diameter of the housing can be considered equal to the outside diameter of the bearing, and the housing length can be set equal to the bearing length.

The length of the shaft is an important parameter in the assembly design and enters into the manufacturing cost as well as the stress and deflection constraints. In either case, the length of the shaft is forced to be the least possible. The cost consideration dictates that the length of the shaft be as low as possible, since a shorter shaft is cheaper to manufacture. Similarly the stress and deflection constraints are better satisfied for a shorter length. If the shaft length is considered as a design variable, the bounds on this variable have to be supplied (the upper bound and the lower bound). Thus it is logical to expect that the optimization process will force the shaft length to the lower bound. Thus it is advantageous to eliminate the length of the shaft from the vector of design variables and allow the user to input the lowest acceptable value as the actual shaft length.

Of the numerous parameters related to the optimization problem of manufacturing and design, some are more important than others. The user may need to use different sets of design variables for different problems. Similarly, the user might be interested in considering one or more parameters to be constant or in specifying relationships between parameters. The mathematical programming formulation developed in this work is capable of providing this flexibility in the selection and specification of design variables and other parameters. It gives the user the freedom to define a design vector that includes only a subset of the large number of relevant parameters. The user can also impose any relationships that are required between any of the parameters. This aspect will be presented in detail in the next chapter.

## 4.2 Design of Sblies :

In order to write a computer code which can handle machine design and manufacturing considerations it is necessary to develop a programmable procedure for analysing a given design. The computer code should be capable of performing all conventional machine design calculations considering the safety factors, the cost factors and the manufacturing considerations. This is achieved by writing subroutines to perform the design calculations and to evaluate related constraint functions.

The American Society of Mechanical Engineers is the sponsor of a code for the Design of Transmission Shafting approved by the American Engineering Standards Committee. This code is based upon the assumption that the shaft is made of a ductile material whose ultimate tensile strength is twice the ultimate shear strength. For this case, the shaft diameter is controlled by the maximum-shear theory regardless of the ratio of the twisting moment to the bending moment. The A.S.M.E. code equation [14] for a hollow shaft subjected to torsion, bending and axial loads is :

$$d_0^3 = \frac{16}{\pi s_s} \left| \left| K_m M + \alpha F_a d_0 (1+K^2)/8 \right|^2 + (K_t T)^2 \right|^{1/2} \times \frac{1}{1-K^4} \qquad 4.1$$

where, $d_0$ = shaft diameter, in.

$F_0$ = axial tension or compression, lb.

$K$ = ratio of inner to outer diameter (=0 for hollow shafts )

$K_m$ = combined shock and fatigue factor to be applied to computed bending moment

$K_t$ = combined shock and fatigue factor to be applied to computed torsional moment

$M$ = maximum bending moment , lb-in.

$T$ = maximum torsional moment, lb-in.

$s_s$ = maximum stress permissible in shear, psi

$\alpha$ = ratio of the maximum intensity of stress resulting from the axial load to the average axial stress.

The value of $\alpha$ is obtained by considering the axial load, or thrust, as a load on a column of diameter d and having a length equal to the distance between the bearings. A straight-line formula commonly used for columns having a slenderness ratio less than 115 gives

$$\alpha = 1/(1-0.0044(L/k)) \qquad 4.2$$

where $L$ = length between supporting bearings, in.

$k$ = radius of gyration of the shaft, in.

Table 4.1 gives the values of working stresses for shafts while Table 4.2 provides the combined shock and fatigue factors.

MAXIMUM PERMISSIBLE WORKING STRESSES FOR SHAFTS

| Grade of shafting | Combined bending and torsion(psi) |
|---|---|
| "Commercial steel" shafting without allowance for keyways... | 8000.00 |
| "Commercial steel" shafting with allowance for keyways........... | 6000.00 |
| Steel purchased under definite specifications.................. | 30% of the elastic limit but not over 18% of the ultimate in tension |

Table 4.1 Maximum Permissible working stresses for shafts

(Reproduced from [14])

| Type of loading | $K_m$ | $K_t$ |
|---|---|---|
| Gradually applied and steady loads | 1.5 | 1.0 |
| Suddenly applied loads with minor shock only | 1.5-2.0 | 1.0-1.5 |
| Suddenly applied loads with heavy shock | 2.0-3.0 | 1.5-3.0 |

$K_m$ = combined shock and fatigue factor to be applied to the computed bending moment

$K_t$ = combined shock and fatigue factor to be applied to the computed torsional moment

Table 4.2 Combined Shock and fatigue factors.

(Reproduced from [14])

In order to use the shaft design equation, the following information is needed :

(i) M, the maximum bending moment lb-in.

(ii) T, the maximum torsional moment lb-in.

(iii) F, the axial force, lb.

(iv) constants $K_m$, $K_t$, $s_s$, $\alpha$ , k as defined in Eq. 4.1

For the calculation of bending moment, an analysis of the loading and end conditions of the shaft has to be performed and the value of the maximum bending moment has to be calculated.

There is also a strong need to limit the deflection of the shaft to be less than a specified value because of the following reasons :

(i) The shaft deflection is to be limited to very small values to avoid the whirling of shafts. Whirling of shafts occurs at critical speeds, which correspond to the speeds at which the number of natural vibrations, or natural frequency, equals the number of revolutions per minute. This usually occurs because of the difference in the location of the center of mass of the rotating disk from the axis of rotation of the shaft. As the deflection increases the eccentricity between the center of mass and axis of rotation increases because of the centrifugal force being developed which tries to throw the load away from the rotating axis. This will be discussed in more detail in the latter half of this chapter.

(ii) The deflection of the shaft at various points disturbs the fits and clearances between the mating parts mounted on the shaft.

(iii)More deflection of the shaft means more deviation of the shaft center line from the mean position; this interferes with the accurate alignment of the shaft together with its loads and may cause problems in the proper functioning of the machinery served by transmission.

Hence, it is necessary to calculate the total maximum deflection of the shaft in addition to the maximum bending moment caused by all of the loads on the shaft.

In order to calculate these values, bending moment and deflection are calculated at a number of points on the shaft and the maximum values are found by comparison. The gravity weight of the loads (gears, pulleys, etc.) and the shaft are neglected because these values are small when compared to the transmission forces (gear forces, tension in the belts and the reaction forces at the bearings). Nevertheless, if the user so wishes, he can add these weights to the vertical components of forces in the input data. The shafts are designed assuming fixed end conditions at the bearings. For this assumption to be valid the length of the bearing should be long enough to ensure near zero slope at the ends. These concerns relating to bearing dimensions will be discussed later in the section on bearing design constraints.

## 4.3 Calculation of Bending Moment and Deflection (Point Load Analyis):

Consider a shaft supported at the ends in sufficiently long bearings to warrant the assumption of fixed end conditions and an intermediate load P acting on the shaft.

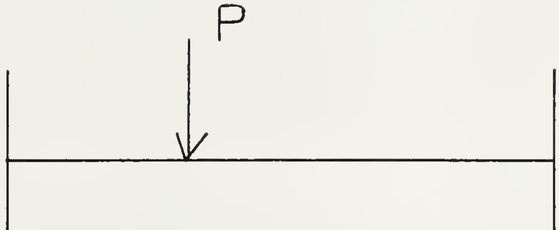



Figure 4.1(a)

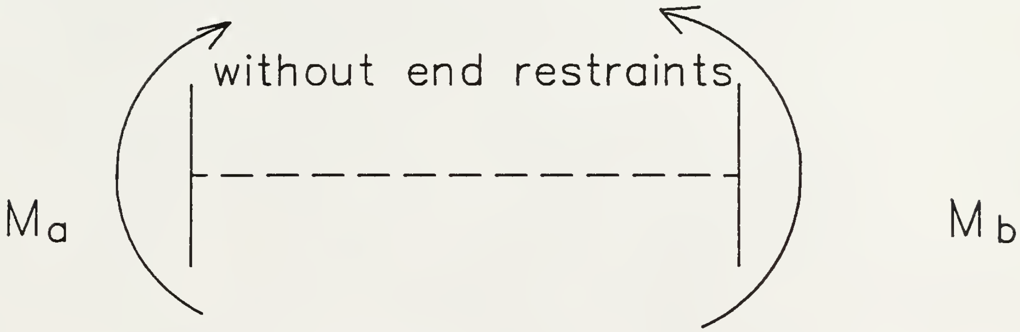



Figure 4.1(b)

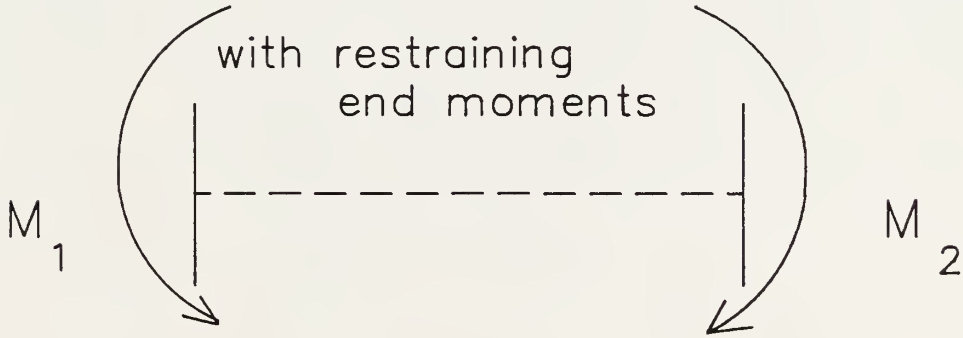



Figure 4.1(c)

The total length between the bearings is L. This can be considered equivalent to the following beam problem as shown in Fig. 4.1(a).

The above fixed-fixed problem can be converted into a simply supported beam problem. A fixed end means that the slope at the end is zero. Because the load P is acting on the shaft it will tend to deform as shown in Fig.4.1 (b) if no moments are applied at the ends.

The load P tends to deflect the beam downwards at every point including the points near the ends. To make the beam have zero slope at the ends, restraining end moments have to be applied as shown in Fig.4.1(c).

These moments should be in the direction that opposes the expected deflection. In the above case the moments should act in the directions shown. Thus the fixed end beam problem can be reduced to a simply supported beam problem with restraining moments $M_1$ and $M_2$ acting at the ends as shown in Fig.4.1(c).

The standard values of $M_1, M_2$ [9] are as follows :

$$M_1 = P\,a\,b^2/L^2 \qquad 4.3$$

$$M_2 = P\,a^2\,b/L^2 \qquad 4.4$$

The positive X-axis is assumed to be directed to the right and the positive Y-axis is directed downwards.

From Shigley [9] we obtain the following results for the case of both ends fixed and an intermediate load acting on the shaft as shown in Fig. 4.1 :

The reactions $R_1$, $R_2$ are :

$$R_1 = Pb^2(3a+b)/L^3 \tag{4.5}$$

$$R_2 = Pa^2(3b-a)/L^3 \tag{4.6}$$

The end moments $M_1$, $M_2$ are:

$$M_1 = Pab^2/L^2 \tag{4.7}$$

$$M_2 = Pa^2b/L^2 \tag{4.8}$$

Bending Moment from A to X :

$$M = -Pab^2/L^2 + R_1x \tag{4.9}$$

Deflection from A to X :

$$y = (Pb^2x^2/6EIL^3)(3ax+bx-3aL) \tag{4.10}$$

Bending Moment from X to B :

$$M = -Pab^2/L^2 + R_1x - P(x-a) \tag{4.11}$$

Deflection from X to B :

$$y = (Pa^2(L-x)^2/6EIL^3)\{(3b+a)(L-x)-3bL\} \tag{4.12}$$

The equations 4.9-4.12 are equivalent to Eqs. 4.13-4.16 .

Bending Moment from A to X :

$$M = -M_1 + R_1x \tag{4.13}$$

Deflection from A to X :

$$y = (Pb^2x^2/6EIL^3)(3ax+bx-3aL) \tag{4.14}$$

Bending Moment from X to B :

$$M = -M_1+R_1x-P(x-a) \qquad 4.15$$

Deflection from X to B :

$$y = (P\ a^2(L-x)^2/6EIL^3)\{\ (3b+a)(L-x)-3bL\} \qquad 4.16$$

If a number of loads $P_1, P_2, P_3$ ... are acting on the shaft, the total bending moment or the total deflection at any section is equal to the sum of the individual bending moments or deflections caused by each load acting alone. This is called the Superposition Principle. Thus we see that the final bending moment and deflection of the shaft can be computed by the above formulas and the superposition principle as follows :

$$BM(i)= BM_1(i)+ BM_2(i)+ BM_3(i) + BM_4(i) \ .......... \qquad 4.17$$

where BM(i) refers to the net bending moment at the section i,

$BM_1$, $BM_2$, etc refer to the bending moment due to each load

$$DEF(i)= DEF_1(i) + DEF_2(i)+ DEF_3(i)+ \ .......... \qquad 4.18$$

where DEF(i) is the net deflection of the shaft at section i,

$DEF_1(i)$, $DEF_2(i)$, etc. are the deflections due to load1, load2, etc.

Once all the values of the bending moment and deflection are computed, the values of the maximum bending moment and maximum deflection of the shaft can be calculated by comparing all the values. The maximum value of the bending moment is used as an input to the shear stress constraint (Eq. 4.1). The maximum deflection constraint forces the maximum deflection value to be within the desired limit as shown below:

$$\text{Max Def}_{act} - \text{Max Def}_{all} \leq 0.0 \tag{4.19}$$

where, Max $\text{Def}_{act}$ is the actual maximum deflection of the shaft,

Max $\text{Def}_{all}$ is maximum allowable deflection of the shaft

The other important design consideration in the design of transmission systems is the range of deviation of the centerline of the shaft. Often it is desired to maintain the centerline of the transmission shaft close to its mean position i.e. the maximum deviation of the shaft should be as small as possible. To limit the value of the maximum deviation of the shaft another constraint is necessary.

The actual deviation of the shaft centerline from the mean position is a function of the tolerances used in the fits between the mating parts. The maximum deviation of the shaft occurs if the shaft diameter is the smallest possible within the given tolerance, the inner diameter of the bearing is the largest possible , the outer diameter of the bearing is the smallest allowable and the housing hole diameter is the largest allowable. This extreme case can be used to formulate the constraint equation for restricting the shaft deviation from the mean position as follows :

$$\text{Max Dev}_{act} - \text{Max Dev}_{allow} \leq 0.0 \tag{4.20}$$

where $Max\ Dev_{act}$ is the actual maximum possible deviation of the shaft centerline,

$Max\ Dev_{allow}$ is the maximum allowable deviation and is user defined.

For the worst case situation explained earlier, the actual deviation of the shaft centerline is given by :

$$Max\ Dev_{act} = ABS(LTOLOS) + UTOLIB + ABS(LTOLOB) + UTOLIH \quad 4.21$$

where, LTOLOS = Lower tolerance on the outside diameter of the shaft

UTOLIB = Upper tolerance on the inside diameter of the bearing

LTOLOB = Lower tolerance on the outside diameter of the bearing

UTOLIH = Upper tolerance on the hole diameter of the housing

Substituting Eq. 4.21 into Eq. 4.20 we get the required constraint as follows :

$$ABS(LTOLOS) + UTOLIB + ABS(LTOLOB) + UTOLIH - Max\ Dev_{allow} \leq 0.0 \quad 4.22$$

where the various parameters are as shown in Eq. 4.20-4.21 .

## 4.4 Constraint to maintain distance between loads :

The loads that act on the shaft affect the bending moment and deflection. The effect of the loads depends not only on the magnitude of the forces but on their distribution along the shaft as well. Thus the distribution of the loads on the shafts should not be decided arbitrarily; rather, the distances of the loads from a fixed reference end of each shaft should be considered as variables included in the design vector. Then the values of these variables will be set by the optimization process. The constraint can be applied to have a minimum fixed proportion of the length of the shaft (say 1/5 of shaft length) between any two loads. The constraint will be of the form :

$$\text{DISLOAD} - \text{SHLEN}/5 \leq 0.0 \qquad 4.23$$

where, DISLOAD is the distance between any two loads or a load and the shaft end ,

SHLEN is the length of the shaft

4.5 Constraint for design of bearings:

The design of bearings is done in accordance with the conventional methods used in engineering practice. Any particular shaft may be supported by two bearings , one at each end. The inside diameter of bearings on the same shaft are the same. Further, the length of the bearing is also considered proportional to the length of the shaft. Hence the design variables of both bearings on a shaft are essentially the same. Thus, during design analysis only one bearing is considered for each shaft.

As per established engineering practice for the fixed end condition of the shaft, the length of the bearing should be at least one tenth of the length of the shaft and the thickness of the bearing is usually taken to be one eighth of the internal diameter of the bearing. Thus two new constraints are to be included for the bearing design considerations. They are of the form :

$$\text{BLEN} - .10 * \text{SHLEN} \leq 0. \qquad 4.24$$

where BLEN is the bearing length,

SHLEN is the shaft length.

$$(\text{DIAOB}-\text{DIAIB})/2 - \text{DIAIB}/8 \leq 0. \qquad 4.25$$

where DIAOB is the outer diameter of the bearing,

DIAIB is the inner diameter of the bearing.

## 4.6 Extension to Multiple Shaft Problems :

The methods developed for solving the single shaft problem can be easily extended to handle the multiple shaft problem with N shafts by repeating the above process N times.

The design vector now contains not only the variables of the first shaft assembly, but also the manufacturing and design variables of the other shafts and their components in the assembly. Thus the variables such as the upper and lower tolerances on the diameter of the shaft (UTOLOS, LTOLOS), the inner diameter of the bearing (DIAIB), the cutting speed for turning the outer surface of the shaft (SPOS), the feed rate for turning the outer surface of the shaft (FDOS), cutting speed for turning inner surface of the bearing (SPIB), etc. have to be treated as arrays to account for multiple shaft analysis. For example, FDOB(i) will now correspond to the cutting feed for the turning operation on the outer surface of the bearings on shaft i. Similarly BLEN(i) denotes the length of the bearing on shaft i. As stated in the preceding section, all bearings on a shaft are assumed to have the same dimensions; similarly the hole diameter of the housing is assumed to be the same for both supporting ends on the shaft. However for obtaining the cost function the number of all similar components have to be taken into consideration since the total cost of manufacture equals the cost of manufacture of all the components in the assembly.

Regarding the application of the constraints, all of the constraints that have been discussed for the single shaft problem have to be applied for every shaft individually in the multiple shaft problem. For every iteration the loads on the corresponding shaft are considered and the corresponding shear stress constraint, maximum deflection constraint, maximum deviation constraint, the bearing design constraints, the load distribution constraints, etc. are applied. These constraints force the corrresponding component design variables to values which satisfy all the constraints besides reducing the cost of manufacture. However some additional constraints have to be applied for the multiple shaft problem. One such type of constraint is applied to maintain the distance between the pairs of shafts as discussed below.

In addition to limiting the centerline deviation of individual shafts, the engineer often faces the problem of ensuring that the distances between different transmission shafts in an assembly are accurately maintained. Errors in these distances occur because of the manufacturing tolerances associated with the various components. The maintainance of accuracy in the distance between the shafts is important because a fluctuation away from the desired mean distance might have an adverse effect on the performance of the system, particularly if meshing gears are mounted on the shafts. Thus the following type of constraint is required :

$$\text{Max Dev } (i,j)_{\text{actual}} - \text{Max Dev } (i,j)_{\text{allow}} \leq 0.0 \qquad 4.27$$

where, Max Dev $(i,j)_{\text{actual}}$ is the maximum possible deviation from the mean distance between the pair of shafts (i,j)

Max Dev $(i,j)_{allow}$ is the maximum allowed deviation from the mean distance between the pair of shafts (i,j)

with i taking values from 1 to N, the number of shafts

j taking values from i+1 to N, the number of shafts.

The term ' Max Dev $(i,j)_{possible}$' is a function of the tolerances on the mating components holding the shaft and is given by the following:

Max Dev$(i,j)_{possible}$ =

[ ABS(LTOLOS(i)) + UTOLIB(i)+ ABS(LTOLOB(i))+ UTOLIH(i)] +

[ ABS(LTOLOS(i))+ UTOLIB(j)+ ABS(LTOLOB(j))+ UTOLIH(j)] 4.28

where the terms LTOLOS, UTOLOS, etc. are same as in Eq. 4.16 and i,j refer to the index numbers of the shafts as described in 4.27 . The maximum deviation allowed is a value that is set by the designer.

Another set of constraints that is necessary to achieve the extension of the single shaft problem to the N-shaft problem is the set of constraints needed to maintain the position of the loads on different shafts. For example if two gears mounted on two different shafts are in mesh, then it is necessary that the distance of these two gears be exactly the same from the fixed reference.

The best way to implement the multiple shaft analysis is to apply the constraints as general equations, with the variables now being arrays and the indices of the arrays corresponding to the index number of the different shafts. The process of application of constraints is looped through N times.

## 4.7 The extended design vector :

In view of the preceding discussion on the integration of shaft design and manufacture within a single optimization problem, the following set of variables are added to the design vector described in chapter III :

DIAOS(I) ------ Diameter of shaft i

DIAIB(I) ------ Inner diameter of bearing on shaft i

DIAOB(I) ------ Outer diameter of bearing on shaft i

BLEN(I) ------ Length of bearing on shaft i

AHDIS(I,J) ------ Distance of horizontal load j on shaft i

AVDIS(I,J) ------ Distance of vertical load j on shaft i

(The last two distances above are measured from a fixed reference end of the shaft)

The manufacturing design variables as already discussed in chapter III are :

FDOS(I) ------- Feed rate for the shaft i outer surface

FDIB(I) ------- Feed rate for the inner surface of bearing i

FDOB(I) ------- Feed rate for the outer surface of bearing i

FDIH(I) ------- Feed rate for the inner surface of housing i

SPOS(I) ------- Cutting Speed for shaft i outer surface

SPIB(I) ------- Cutting Speed for inner surface of bearing i

SPOB(I) ------- Cutting Speed for outer surface of bearing i

SPIH(I) ------- Cutting Speed for inner surface of housing i

UTOLOS(I) ------- Upper tolerance for shaft i outer surface

LTOLOS(I) ------- Lower tolerance for shaft i outer surface

UTOLIB(I) ------- Upper tolerance for inner surface of bearing i

LTOLIB(I) ------- Lower tolerance for inner surface of bearing i

UTOLOB(I) ------- Upper tolerance for outer surface of bearing i

LTOLOB(I) ------- Lower tolerance for outer surface of bearing i

UTOLIH(I) ------- Upper tolerance for hole in housing i

LTOLIH(I) ------- Lower tolerance for hole in housing i

In the above definitions i takes the values from 1 to N (the number of shafts), and

j takes the values from 1 to the number of loads.

The manufacturing parameters that are involved in the optimization problem indirectly are :

(i) Surface finish of all component inner and outer surfaces ,

(ii) Nose radius of tool for turning the various surfaces of components.

These parameters affect the design through constraint functions as shown in chapter II.

In the next chapter the implementation of these concepts in a computer code is discussed and the optimization techniques used are described in detail.

# CHAPTER V

## OPTIMIZATION TECHNIQUES AND IMPLEMENTATION

### 5.1 Introduction

Optimization techniques are of great value in engineering design. The traditional design process makes extensive use of empirical charts, tables, formulas and procedures developed through many years of experience. Optimization methods, on the other hand, are based on the idea of applying established numerical techniques to reasonable mathematical models of the system to be designed. A computer code which implements such an optimization method will be capable of analyzing the proposed design and forcing the various parameters towards an optimal solution in order to satisfy the desired requirements.

In general, the design problem is reduced to the following form:

$$\text{Minimize } F(B) \qquad 5.1$$

$$\text{subject to } g_j(B) \leq 0 \qquad j=1,m \qquad 5.2$$

$$g_k(B) = 0 \qquad k=m+1,n \qquad 5.3$$

where, B is the vector of design variables $[b_1, b_2, b_3, b_4 \ldots b_{nv}]^T$

m is the number of inequality constraints and

n is the total number of constraints.

For the minimization of the function F(B) there are a number of optimization techniques that are readily applicable. These techniques are usually iterative in nature and produce an improved design at each iteration until the process converges to the optimum. A proper direction is selected at every iteration, in order to move towards the optimum. The value of the function is calculated and the design point is updated at every iteration. The use of optimization techniques yields a result that is the best design in some particular sense whereas conventional design merely provides an acceptable design. Further, this approach requires a lower level of skill on the part of the designer. Hence the optimal design approach is well suited for design automation and design-manufacture integration. The disadvantages of this approach include large computing requirements and the difficulty associated with the translation from an engineering design problem to an optimal design problem.

This chapter covers the basic optimization techniques related to the work presented in this thesis, along with a discussion of the optimization routine actually used in the computer code. A detailed explanation of the code and its usage is also given.

## 5.2 Optimization Techniques

For problems involving cost and constraint functions whose derivative evaluations are complicated, as in the case of this work, it is advisable to use a non-derivative optimization technique. These techniques are based upon function evaluations only at each iteration and, unlike derivative based methods, there is no need for derivative evaluations in order to establish a suitable direction of descent. Some of the optimization techniques that are most popular among derivative-free methods are grid search [8], random search[8] and Hooke Jeeves [8] methods. A routine called MINA, which is based upon the grid search method, was developed by Sandia Laboratories as part of the Sandia Mathematical Subroutine Library. This routine is reliable, fairly efficient and easy to use; hence, it is suitable for complicated problems with many variables and constraints.

MINA finds an approximate minimum of a real-valued function of NV variables, given an initial estimate of the position of the minimum and the ranges for each of the variables. This routine uses a selective directed search of a surrounding NV-dimensional grid of points to find a direction in which the function decreases. It then proceeds in this direction as far as the function decreases, then determines a new search direction. When no such direction is found the step size is decreased and the process is repeated.

To ensure complete satisfaction of design requirements, a number of constraints have to be imposed on the optimization problem. The routine MINA is basically an unconstrained optimization routine. To account for the constraints an exterior penalty function method [8] is used, as explained below.

Referring to Eq. 5.1 through 5.3, the function F(B) is to be minimized subject to the given set of constraints. At every function evaluation the constraint violation for all design and manufacture requirements is checked. If any constraints are violated, the design is not totally feasible. In such cases the exterior penalty function method that is employed adds a penalty factor to the true value of the cost function to generate a pseudo-objective function which is given by the following equation :

$$POF = FF + PP \qquad 5.4$$

where, POF is the pseudo-objective function,

FF is the true value of the cost function and

PP is the penalty factor defined by :

$$PP = R_p * GG \qquad 5.5$$

where $R_p$ is a constant parameter defined by user , and

GG is the summation of all the constraint violations.

The optimization routine manipulates the values of the design variables within their respective ranges (the range on each design variable is defined by the user) so as to reduce the value of the pseudo-objective function. The more the constraint violation, the greater is the value of the pseudo-objective function. Hence, unconstrained minimization of the pseudo objective function takes the design vector closer to the feasible region (i.e. the constraint violation is decreased) at each iteration. Within the feasible region, the penalty is zero and the true cost function and the pseudo-objective function are identical. Thus, minimizing the pseudo-objective function also minimizes the true cost function. The process is considered to have converged if all the constraints are satisfied and no significant cost reduction is observed for a full iteration.

Otherwise the process is repeated with all parameters being updated after every iteration.

## 5.3 Development of the Computer Code

The concept of integration of design and manufacture into a single optimization problem which can be solved using the grid search optimization technique was implemented in an efficient computer code. The structure of the code is as shown in Fig. 5.1 . In the following paragraphs the description of the various routines used in the code is provided.

Subroutine MINA:-

As discussed in section 5.2 MINA is a grid search based optimizing routine. The following are the details regarding the input and output parameters of the routine.

Input to subroutine MINA :

FN ... Name of the function of NV variables to be minimized. This name must appear in an external statement. The form of the calling sequence must be FUNCTION FN(B), where B is an array of NV variables. The function name used in the current implementation is SUBM.

NV.... Number of variables

NDIV.. Number of refinements of the search increments to use. At each refinement, the increment in each dimension is divided by 10. (Generally NDIV is taken as 3 or 4).

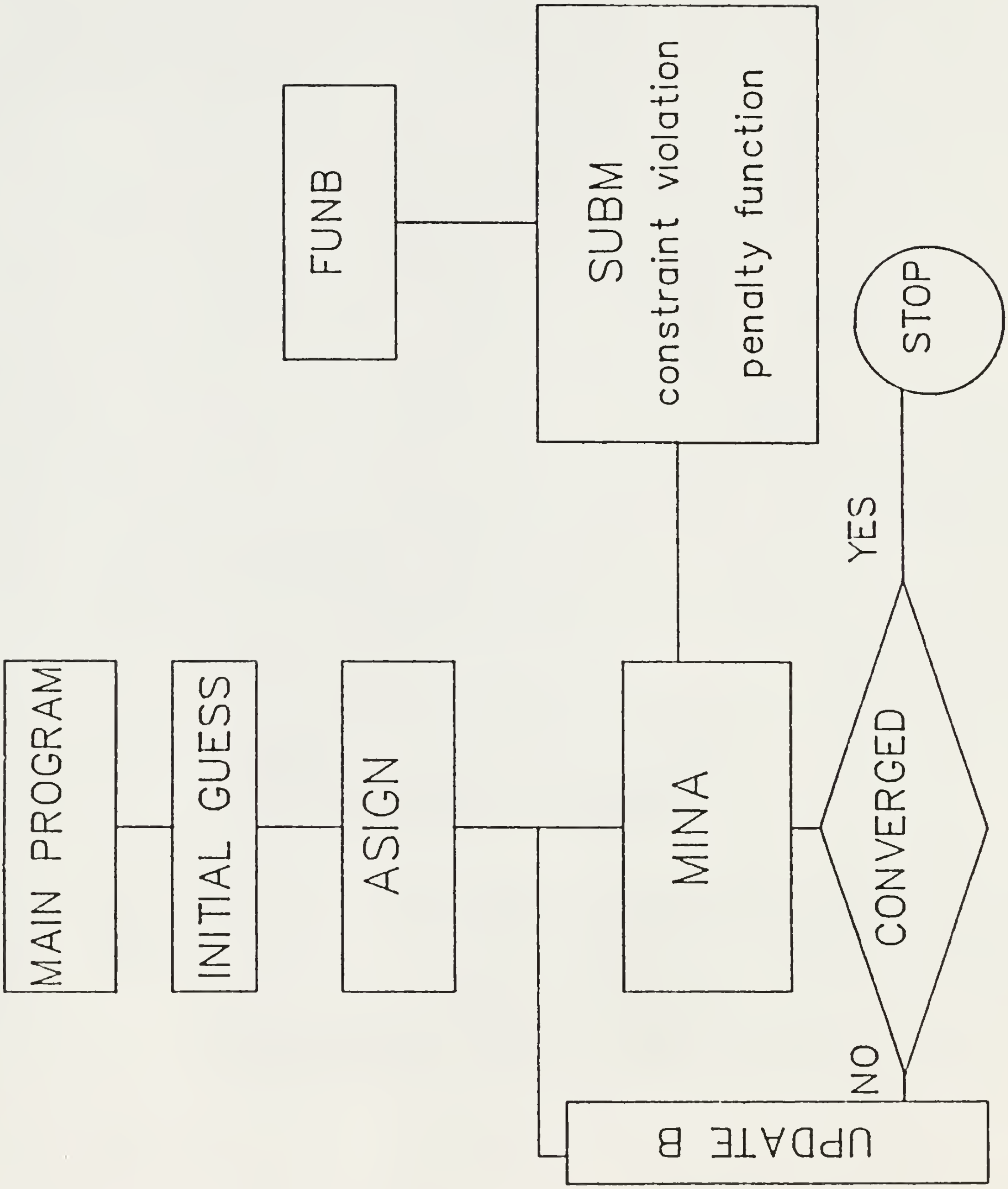

MAIN PROGRAM
INITIAL GUESS
ASIGN
MINA
FUNB
SUBM
constraint violation
penalty function
CONVERGED
YES
NO
STOP
UPDATE B

Fig 5.1

DEL... Fraction of variable range (in each dimension) to use as the initial increment (in that dimension).

A.... Array of search bounds, dimensioned NV × 2

A(I,1) should be the lower bound of the i-th variable

A(I,2) should be the upper bound of the i-th variable

GUESS.. Array of NV initial values. GUESS(I) should be the initial value for the i-th variable.

The output returned by subroutine MINA :

X.... Array (dimensioned NV) giving the values of the variables at the minimum. X(I) will be the value of the i-th variable.

FOFX... Function value at the minimum

IERR... A status code

-----Normal Code

=1 Means the search for a minimum proceeded for the specified number of refinements.

-----Abnormal Code

=2 Means NV is greater than 60

=3 Means a range minimum is greater than the corresponding maximum.

Function Subprogram SUBM:

SUBM is a function subprogram which is repeatedly called by the routine MINA. This subprogram calls a subroutine FUNB which returns the true cost function. Within SUBM all the constraint functions are evaluated. The cost and constraint function values are then combined into the pseudo-objective function which is then returned to MINA.

Subroutine ASSIGN :

This is the routine that the user is required to provide to the program to define the set of design variables. This routine establishes the correspondence between the various design variables and the manufacturing and design parameters. Through this subroutine, the user has direct control over the design vector and can designate as design variables only those parameters which are significant for the particular problem at hand. Hence, for problems of a particular class, the parameters not seriously affecting the design can be set to constant values instead of treating them as design variables. This results in a smaller optimization problem that can be solved more efficiently. The user can also use this subroutine to enforce constraints between the various parameters.

For instance, considering the example shown in Fig.3.1, for the design and manufacture of a single shaft subjected to point loads and supported by fixed bearings at the ends, the following is the set of statements needed to define the design vector and set the values of other parameters. It may be noted that some of the parameters are equated to constant values as per the discussion in chapter IV.

SPOS(1) = B(1) ... Cutting speed for shaft

SPIB(1) = B(2) ... Cutting speed for turning inner surface of bearing

SPOB(1) = B(3) ... Cutting speed for turning outer surface of bearing

SPIH(1) = B(4) ... Cutting speed for the housing hole

```
FDOS(1) = B(5)    ... Cutting feed rate for shaft surface

FDIB(1) = B(6)    ... Cutting feed rate for inner side of bearing

FDOB(1) = B(7)    ... Cutting feed rate for outer side of bearing

FDIH(1) = B(8)    ... Cutting feed rate for housing hole surface

UTOLOS(1) = B(9) ... Upper tolerance for shaft diameter

LTOLOS(1) = B(10)... Lower tolerance for shaft diameter

UTOLIB(1) = B(11) .. Upper tolerance for inner diameter of
                     bearing

LTOLIB(1) = B(12) .. Lower tolerance for inner diameter of
                     bearing

UTOLIH(1) = B(13) .. Upper tolerance for hole diameter of
                     housing

LTOLIH(1) = B(14) .. Lower tolerance for hole diameter of
                     housing

SLEN(1)   = 50.0  .. Shaft length being considered as constant

BLEN(1)   = B(15) .. Bearing length

HLEN(1) = BLEN(1) .. Housing length set equal to bearing length

DIAOS(1)  = B(17) .. Outer Diameter of shaft

DIAIB(1)=DIAOS(1) .. Inner diameter of bearing set equal to
                     outer diameter of shaft

DIAOB(1)  = B(18) .. Outer diameter of shaft

DIAIH(1)=DIAOB(1) .. Diameter of housing hole set equal to outer
                     diameter of bearing

AHDIS(1,1) = B(19) .. Distance of first horizontal load on
                      shaft from reference end
```

AHDIS(1,2) = B(20) .. Distance of second horizontal load on shaft from fixed reference end

AHDIS(1,3) = B(21) .. Distance of third horizontal load on shaft from fixed reference end

Similar input is required for all the components if the number of shafts is greater than one. The only difference is that the subscript in each of the variables is set to the index number of the shaft being described.

Subroutine INITIAL-GUESS :

This subroutine reads input data from an input file.The input file contains the initial guess values for all the design variables besides the lower and upper bounds on each of these variables. The file also contains values of other necessary constants like moduli of elasticity of the materials, the shear stress limits of the materials, the desired values of surface finish, tolerances on dimensions, maximum allowable deflections of shaft(s), maximum allowable deviationsof the shafts, etc. The loading of the various shafts is also read from this input file. In order to change some of the input, only the data file needs to be changed. If an entirely different problem is to be solved then the ASSIGN routine needs to be changed in addition to modifying the input file.

This routine also echoes all the input values to an output file, so that the user can check that the values being read by the code are correct.

The optimal design code developed is easy to use, efficient and reliable. For a given problem, it yields a set of design and machining parameters that satisfy all design and manufacturing criteria besides ensuring minimum possible cost for a feasible design. Further the user has complete freedom to try any choice of variables, constraints and material properties to arrive at a design that best meets the particular needs of the problem at hand.

# CHAPTER VI

## NUMERICAL EXAMPLES

This chapter discusses the use of optimal design methodology and the computer code described in this thesis for solving actual design and manufacturing problems. Several numerical examples are presented along with the results that were obtained. These numerical examples were formulated considering various machine design problems from different sources [9,13,14]. In the first two examples the design of the assembly is fixed and only the manufacture is optimized. The next three examples demonstrate how the design and manufacture can be simultaneously optimized using the proposed method.

### 6.1 Single shaft transmission assembly manufacture :

(a) A single shaft is to be manufactured along with two bearings, each one supporting the shaft at one end and embedded in a housing hole. The various parts of the transmission assembly as shown in Fig. 6.1 are to be manufactured by the turning process. The optimal set of machining parameters needs to be found for the case of minimum cost of manufacture. The following are the part specifications :

Surface Finish on the shaft surface : 250 micro in. (rms)

Surface Finish on the bearing hole surface : 200 micro in. (rms)

Surface Finish on the bearing outer surface: 325 micro in. (rms)

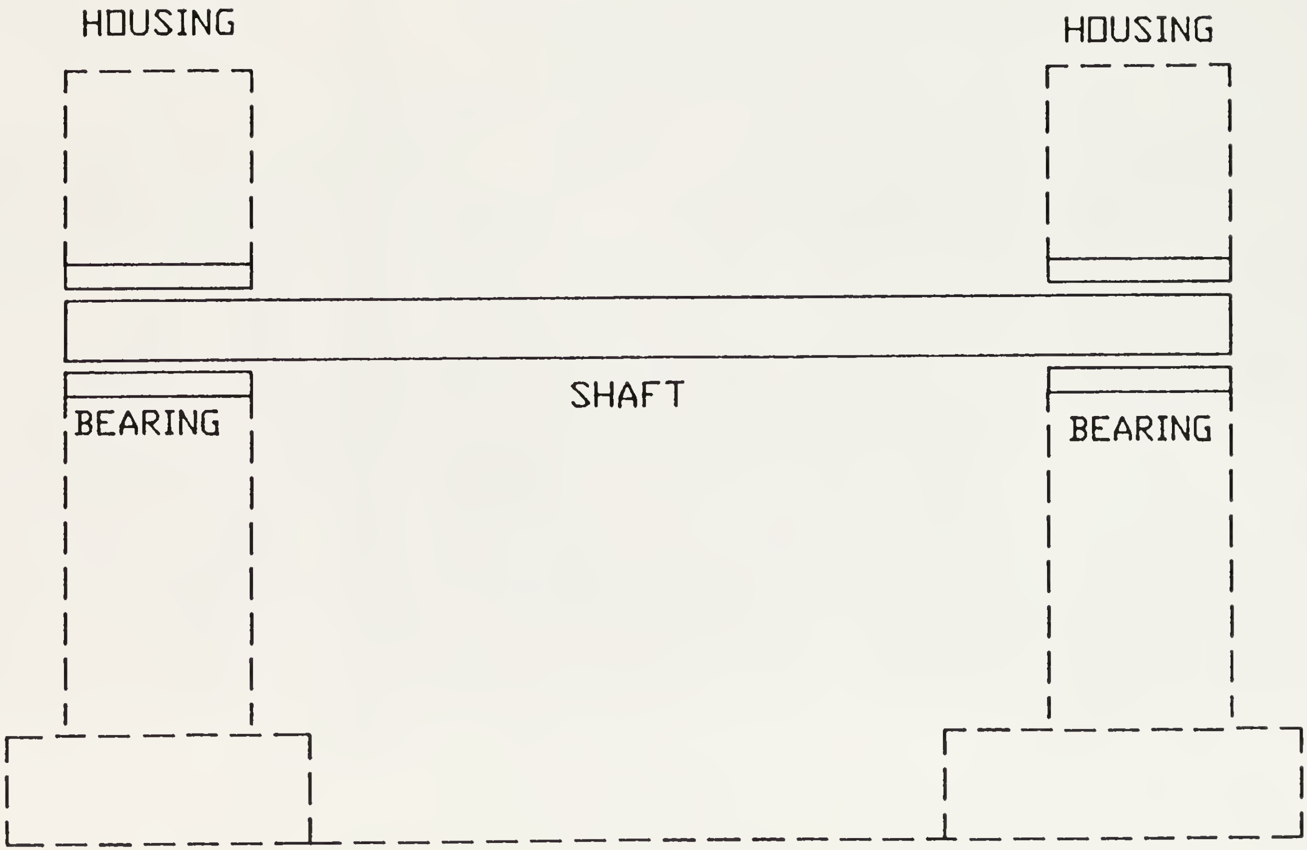



Figure 6.1 Components of single shaft transmission assembly

| | |
|---|---|
| Surface Finish on the housing hole | : 300 micro in. (rms) |
| Minimum tolerance gap between the upper and lower tolerance | : 0.00155 in. |
| Maximum clearance between the shaft outer surface and the inner surface of bearing | : 0.010 in. |
| Minimum clearance between the shaft outer surface and the inner surface of bearing | : 0.0057 in. |
| Maximum clearance between the bearing outer surface and housing hole | : 0.0057 in. |
| Minimum clearance between the bearing outer surface and housing hole | : 0.002 in. |
| Nominal cutting feed | : 0.04 ipr |
| Maximum tool nose radius | : 0.6 in. |
| Diameter of the shaft | : 3.0 in. |
| Length of the shaft | : 50.0 in. |

The results obtained are recorded in Table 6.1

RESULTS FOR EXAMPLE 6.1(a)

| DESIGN VARIABLE | INITIAL GUESS | FINAL VALUE |
|---|---|---|
| SPOS(1) | 100.0 sfpm | 150.94 sfpm |
| SPIB(1) | 100.0 sfpm | 289.48 sfpm |
| SPOB(1) | 100.0 sfpm | 289.48 sfpm |
| SPIH(1) | 100.0 sfpm | 150.67 sfpm |
| FDOS(1) | 0.03 ipr | 0.0195 ipr |
| FDIB(1) | 0.04 ipr | 0.0195 ipr |
| FDOB(1) | 0.04 ipr | 0.0144 ipr |
| FDIH(1) | 0.03 ipr | 0.0400 ipr |
| UTOLOS(1) | -0.003 in | -0.0026 in |
| LTOLOS(1) | -0.0045 in | -0.00475 in |
| UTOLIB(1) | 0.005 in | 0.00525 in |
| LTOLIB(1) | 0.0035 in | 0.0031 in |
| UTOLOB(1) | -0.0035 in | -0.001 in |
| LTOLOB(1) | -0.0045 in | -0.0028 in |
| UTOLIH(1) | 0.005 in | 0.0028 in |
| LTOLIH(1) | 0.003 in | 0.0010 in |

Initial value of Pseudo Objective Function = 117709.44

Final value of Pseudo Objective Function = 1.42

Table 6.1

| | | |
|---|---|---|
| Initial value of true cost | = | 1.39 $ |
| Final value of true cost | = | 1.42 $ |
| Initial Constraint Violation | = | 11.770806 |
| Final Constraint Violation | = | 1.250 E-9 |

Table 6.1 (CONTD.)

(b) The same problem is now considered for more demanding surface finish constraints. The following are the part requirements :

| | | |
|---|---|---|
| Surface Finish on shaft outer surface | : | 50.0 micro in. (rms) |
| Surface Finish on inner surface of bearing | : | 80.0 micro in. (rms) |
| Surface Finish on outer surface of bearing | : | 59.0 micro in. (rms) |
| Surface Finish on housing hole surface | : | 89.0 micro in. (rms) |
| Minimum tolerance gap | : | 0.00155 in |
| Maximum clearance between shaft outer surface and bearing inner surface | : | 0.010 in |
| Minimum clearance between shaft outer surface and bearing inner surface | : | 0.0057 in |
| Maximum clearance between bearing outer surface and housing hole surface | : | 0.0057 in |
| Minimum clearance between bearing outer surface and housing hole surface | : | 0.002 in |
| Nominal feed | : | 0.04 ipr |
| Maximum tool nose radius | : | 0.6 in |
| Diameter of shaft | : | 3.0 in |
| Length of shaft | : | 50.0 in |

The results obtained for this example are shown in Table 6.2

RESULTS FOR EXAMPLE 6.1(b)

| DESIGN VARIABLE | INITIAL GUESS | FINAL VALUE |
|---|---|---|
| SPOS(1) | 100.0 sfpm | 150.94 sfpm |
| SPIB(1) | 100.0 sfpm | 289.48 sfpm |
| SPOB(1) | 100.0 sfpm | 289.48 sfpm |
| SPIH(1) | 100.0 sfpm | 151.09 sfpm |
| FDOS(1) | 0.03 ipr | 0.0168 ipr |
| FDIB(1) | 0.04 ipr | 0.0209 ipr |
| FDOB(1) | 0.04 ipr | 0.0126 ipr |
| FDIH(1) | 0.03 ipr | 0.0280 ipr |
| UTOLOS(1) | -0.003 in | -0.00255 in |
| LTOLOS(1) | -0.0045 in | -0.00440 in |
| UTOLIB(1) | 0.005 in | 0.00555 in |
| LTOLIB(1) | 0.0035 in | 0.00325 in |
| UTOLOB(1) | -0.0035 in | -0.0014 in |
| LTOLOB(1) | -0.0045 in | -0.0031 in |
| UTOLIH(1) | 0.005 in | 0.00257 in |
| LTOLIH(1) | 0.003 in | 0.0010 in |

Initial value of Pseudo Objective Function = 3837619.09

Final value of Pseudo Objective Function = 1.65

Table 6.2

| | | |
|---|---|---|
| Initial value of true cost | = | 1.39 $ |
| Final value of true cost | = | 1.65 $ |
| Sum of initial constraint violations | = | 383.761770 |
| Sum of final constraint violations | = | 1.87512 E -09 |

Table 6.2 (CONTD.)

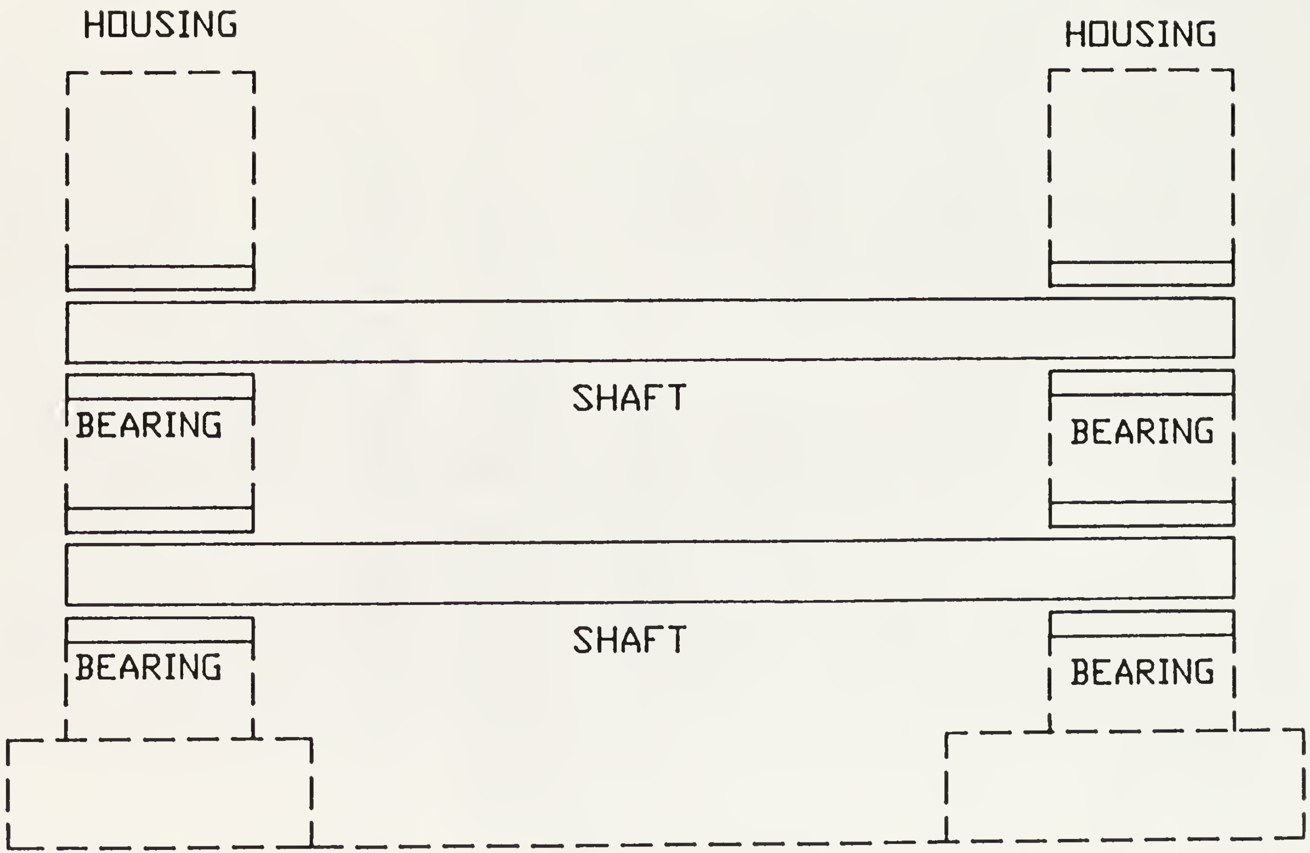



Figure 6.2 Components of double shaft transmission assembly

## 6.2 Double shaft transmission assembly manufacture :

A transmission shaft assembly with two shafts as shown in Fig. 6.2 needs to be manufactured. The various surfaces of the assembly (shaft outer surface, bearing inner surface, bearing outer surface, surface of housing hole) are to be turned to the required dimensions through finish cuts. The optimal set of machining parameters needs to be found for the case of minimum cost of manufacture. The following are the required specifications :

| | |
|---|---|
| Surface finish of outer surface of shafts | : 50.0 mi.in. (rms) |
| Surface finish of inner surface of bearing | : 80.0 mi.in. (rms) |
| Surface finish of outer surface of bearings | : 59.0 mi.in. (rms) |
| Surface finish of inner surface of housing | : 89.0 mi.in. (rms) |
| Maximum clearance between shaft outer surface and inner surface of bearing | : 0.010 in. |
| Minimum clearance between shaft outer surface and inner surface of bearing | : 0.0057 in. |
| Maximum clearance between bearing outer surface and housing hole | : 0.0057 in. |
| Minimum clearance between bearing outer surface and housing hole | : 0.002 in. |
| Minimum gap between tolerance values | : 0.00155 in. |
| Nominal feeds for the turning of components | : 0.04 in. |
| Maximum tool nose radius | : 0.6 in |

| | |
|---|---|
| Shaft length | : 50.0 in |
| Diameter of shaft | : 3.5 in |

The results obtained for this problem are shown in Table 6.3

RESULTS FOR EXAMPLE 6.2

| DESIGN VARIABLE | INITIAL GUESS | FINAL VALUE |
|---|---|---|
| SPOS(1) | 100.0 sfpm | 150.94 sfpm |
| SPIB(1) | 100.0 sfpm | 289.48 sfpm |
| SPOB(1) | 100.0 sfpm | 289.48 sfpm |
| SPIH(1) | 100.0 sfpm | 151.10 sfpm |
| FDOS(1) | 0.03 ipr | 0.0168 ipr |
| FDIB(1) | 0.04 ipr | 0.0209 ipr |
| FDOB(1) | 0.04 ipr | 0.0124 ipr |
| FDIH(1) | 0.03 ipr | 0.0280 ipr |
| UTOLOS(1) | -0.003 in | -0.00255 in |
| LTOLOS(1) | -0.0045 in | -0.00440 in |
| UTOLIB(1) | 0.005 in | 0.00555 in |
| LTOLIB(1) | 0.0035 in | 0.00325 in |
| UTOLOB(1) | -0.0035 in | -0.00140 in |
| LTOLOB(1) | -0.0045 in | -0.00310 in |
| UTOLIH(1) | 0.005 in | 0.00255 in |
| LTOLIH(1) | 0.003 in | 0.00100 in |
| SPOS(2) | 100.0 sfpm | 150.94 sfpm |
| SPIB(2) | 100.0 sfpm | 291.04 sfpm |
| SPOB(2) | 100.0 sfpm | 291.04 sfpm |

Table 6.3

| | | |
|---|---|---|
| SPIH(1) | 100.0 sfpm | 151.10 sfpm |
| FDOS(2) | 0.03 ipr | 0.0187 ipr |
| FDIB(2) | 0.04 ipr | 0.0171 ipr |
| FDOB(2) | 0.04 ipr | 0.0113 ipr |
| FDIH(2) | 0.03 ipr | 0.0280 ipr |
| UTOLOS(2) | -0.003 in | -0.00265 in |
| LTOLOS(2) | -0.0045 in | -0.00490 in |
| UTOLIB(2) | 0.005 in | 0.00510 in |
| LTOLIB(2) | 0.0035 in | 0.00305 in |
| UTOLOB(2) | -0.0035 in | -0.00140 in |
| LTOLOB(2) | -0.0045 in | -0.00310 in |
| UTOLIH(2) | 0.005 in | 0.00255 in |
| LTOLIH(2) | 0.003 in | 0.00100 in |

---

| | | |
|---|---|---|
| Initial value of Pseudo Objective Function | = | 7675246.93 |
| Final value of Pseudo Objective Function | = | 3.50 |
| Initial value of true cost | = | 3.00 $ |
| Final value of true cost | = | 3.50 $ |
| Sum of Initial Constraint Violations | = | 767.524393 |
| Sum of Final Constraint Violations | = | 0.699 E-8 |

---

Table 6.3 (CONTD.)

## 6.3 Single shaft assembly design and manufacture :

A single transmission shaft assembly need to be designed as shown in Fig. 6.3 . The various components of the assembly ( the shaft, bearings and the housing ) are to be manufactured by the turning process. The optimal set of design variables (machining parameters and component design parameters) needs to be found for the case of minimum cost of manufacture using the optimal design code developed in the earlier chapters. The following information gives the requirements and the specifications. Figures 6.3 and 6.4 show the details of this problem.

Pulley B (24 in. diameter) receives 30 hp at 360 rpm from below at an angle of 45 degrees as shown in Fig. 6.4 . The 8 inch gear C delivers 40% of the power horizontally to the right. The 12 in. gear E delivers the remaining power upward towards the right at an angle of 30 degrees above the horizontal. Both gears have 20 degrees involute teeth. The following are the shaft design specifications :

| | | |
|---|---|---|
| Max shaft deflection | : | 6.2 E-3 in. |
| Max shaft deviation | : | 2.75 E-2 in. |
| Max shear stress | : | 29.0 E 6 $lb/in^2$ |

The equation to calculate the torque being transmitted [13] is

Torque = 63000.0*(Horse Power)/(RPM)

Using the above equation, the torque on the shaft between B and C is

$T_b$=5250 in-lb

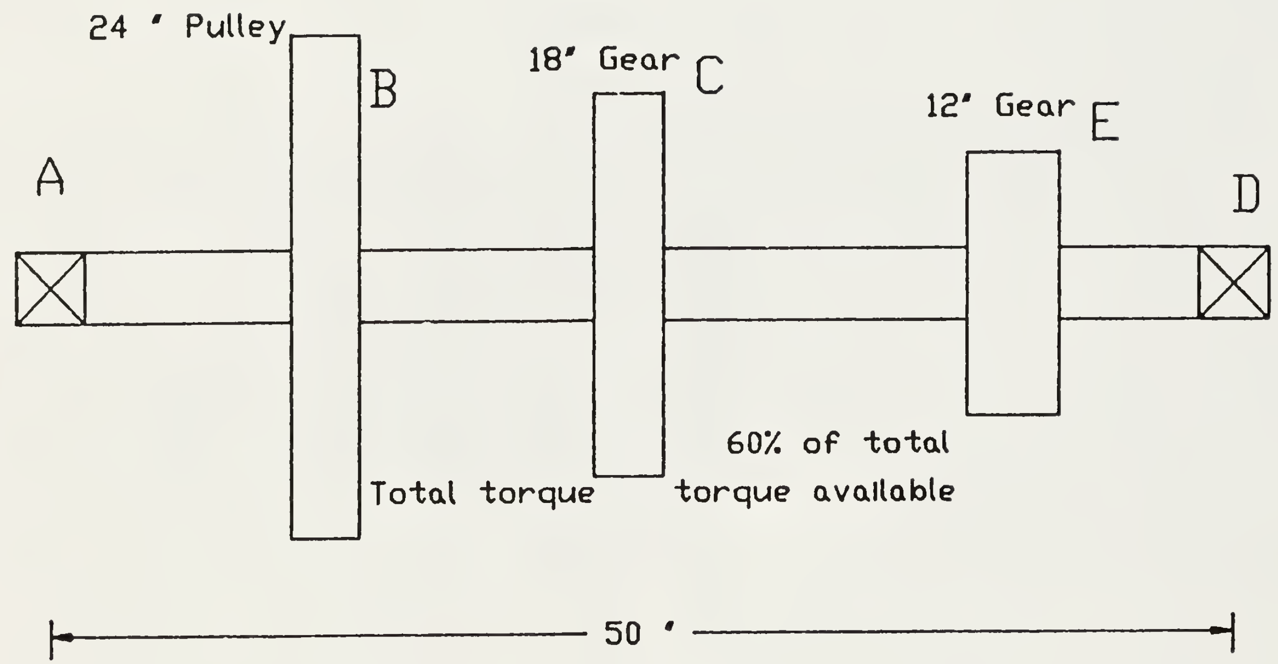



Figure 6.3 Single shaft assembly with transmission loads

Similarly torques at C and E can be calculated as follows

$T_c = 2100$ in-lb

$T_e = 3150$ in-lb

The maximum torque moment MXTRQ acting on the shaft is the input torque $T_b$; thus

MXTRQ = $T_b$ = 5250 in-lb

The bending force produced by the belt is given by the equation

$$F = 2(F_1 - F_2)$$

$$= 2(T_b)/(r_b)$$

where $T_b$ is the torque and $r_b$ is the pitch radius as shown in Fig. 6.4 . Thus bending forces at B, E and C are as follows :

$F_b = 875$ lb.

$F_e = 525$ lb.

$F_c = 233$ lb.

The total force on the gear tooth (ignoring the frictional force) is normal to the tooth surface, with the result that there is a separating force N (as in Fig. 6.5 ), given by

$$N = F \tan\phi,$$

where F is the computed driving force.

For $\phi = 20°$ , the separating forces for C and E are :

$$N_c = F_c \tan 20° = (233)(0.364) = 84.8 \text{ lb.}$$

$N_e = F_e \tan 20° = 191$ lb.

If C delivers power to the right, the force $F_c$ on C is directed to the left, as shown in the end view of Fig. 6.4 . Similarly with E delivering power as stated, the force $F_e$ is upward toward the right. By analytic mechanics, those forces acting at some distance from the center of the shaft are replaced by a force through the shaft axis and a couple. Thus, we can add and subtract forces $F_e$ through the shaft axis as indicated. Now there will be a counter clockwise torsional couple $F_e*r_e$ , where $r_e$ is the pitch radius of the gear E , and $F_e$ is a bending force acting at the center of the shaft parallel to the original $F_e$. This is the basis for the free bodies to be used later.

Now, resolving forces into two perpendicular coplanar systems, the horizontal forces at B, C and E are :

$B_x = F_b* \cos45 =(875)*(0.707) = 619$ lb.

$C_x = F_c = 233$ lb.

$E_x = F_e* \cos30 - N_e*\cos60 = 359.1$ lb.

Thus the horizontal components are as shown in Fig. 6.6(a).

The forces in the vertical plane can be obtained as :

$B_y = F_b*\cos45 = 619$ lb.

$C_y = N_c = 84.8$ lb.

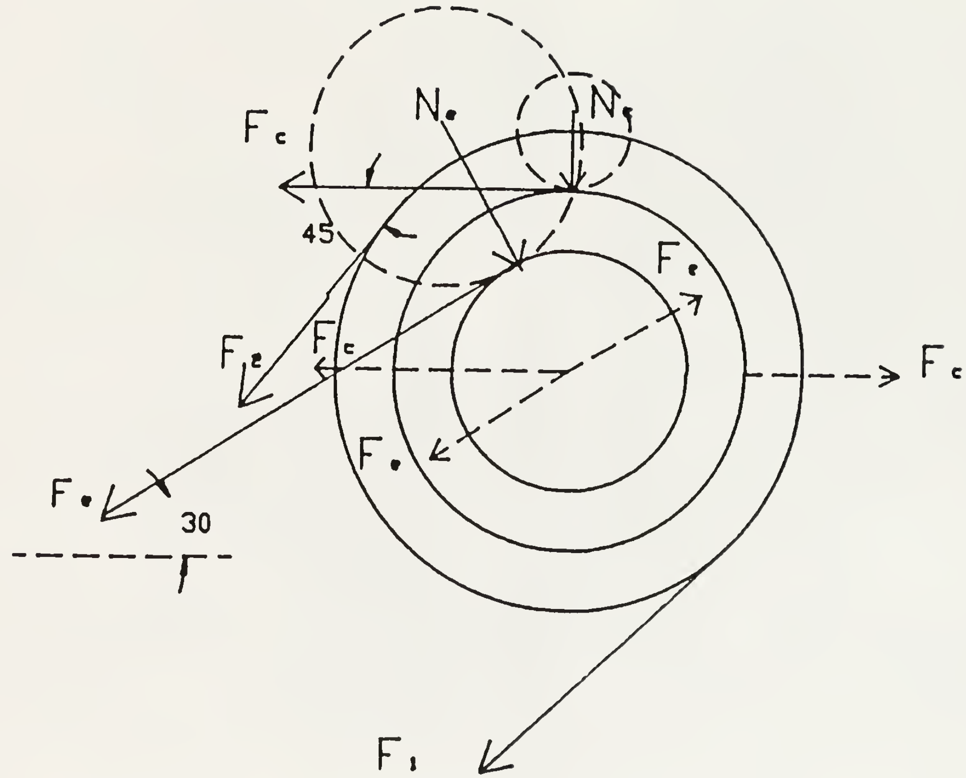



Figure 6.4 Force diagram for the single shaft assembly of Fig 6.3

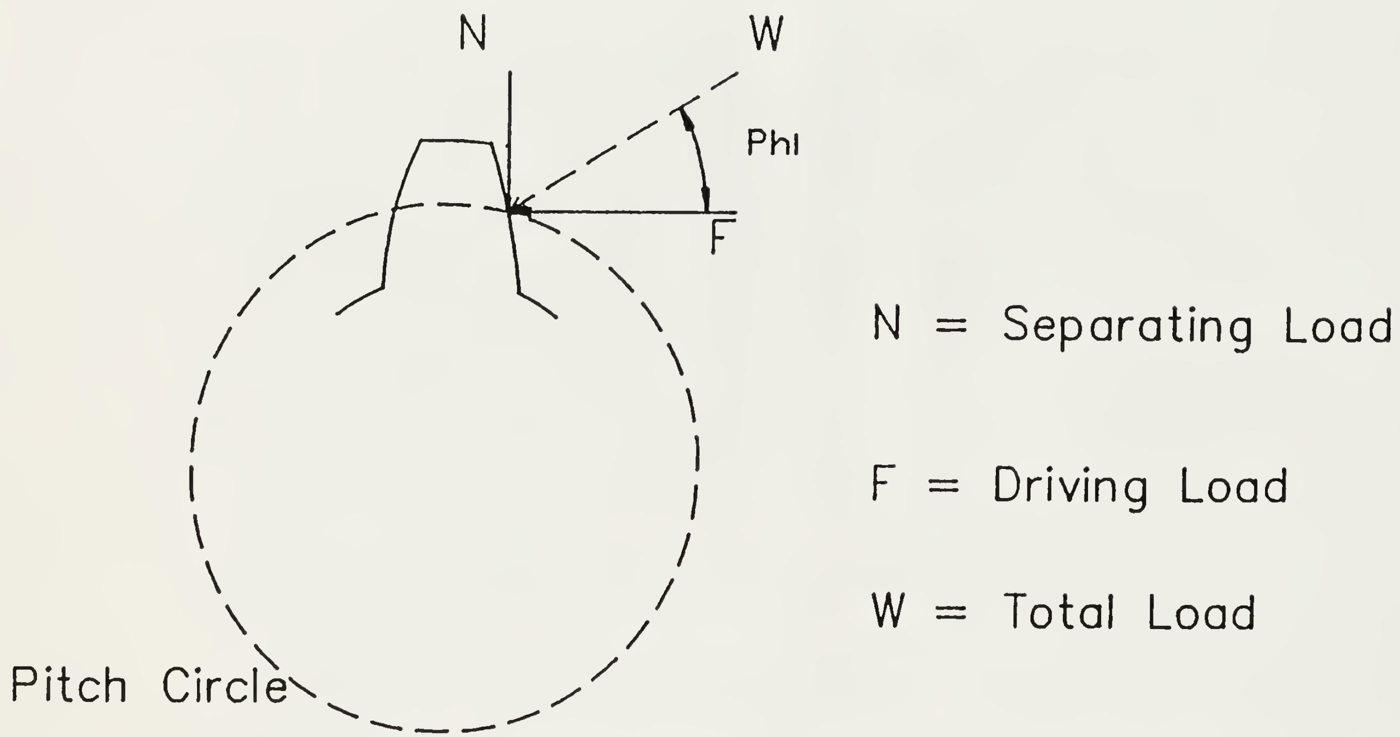



Figure 6.5 Forces on gear tooth

$$E_y = F_e * \sin 30 + N_e * \cos 30 = 427.9 \text{ lb.}$$

Therefore as per the notations discussed in chapter 5 the loading can be described as follows :

HPLOD(1,1) = 619 lb.

HPLOD(1,2) = 233 lb.

HPLOD(1,3) = 359.1 lb.

whereas the vertical loading is as follows :

VPLOD(1,1) = 619 lb.

VPLOD(1,2) = 84.8 lb.

VPLOD(1,3) = 427.9 lb

The vertical loads on the shaft are shown in Fig. 6.6(b) .

These values are substituted into the equations developed in chapter II to obtain the necessary set of design constraints for this problem. The tolerance and surface finish constraints imposed are similar to those in Example 1.

The design variables for this problem will be as follows :

(i) The cutting speeds for the turning of the surfaces of the components,

(ii) The cutting feed rates for the turning of component surfaces

(iii) The manufacturing tolerances on the components.

In addition to the above set of design variables the following shaft design parameters are also included in the design vector :

(i)The diameter of the shaft

(ii)The outer diameter of the bearing

(iii)The length of the bearing

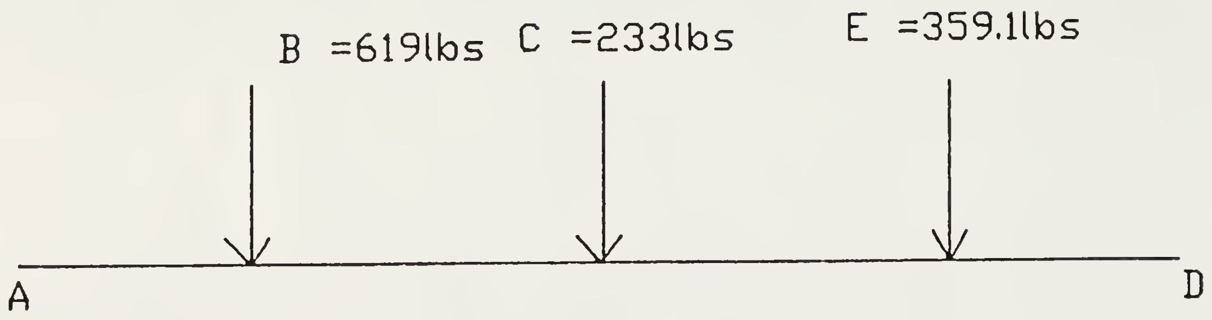



Figure 6.6(a) Horizontal Load Diagram for single shaft of Fig.6.3

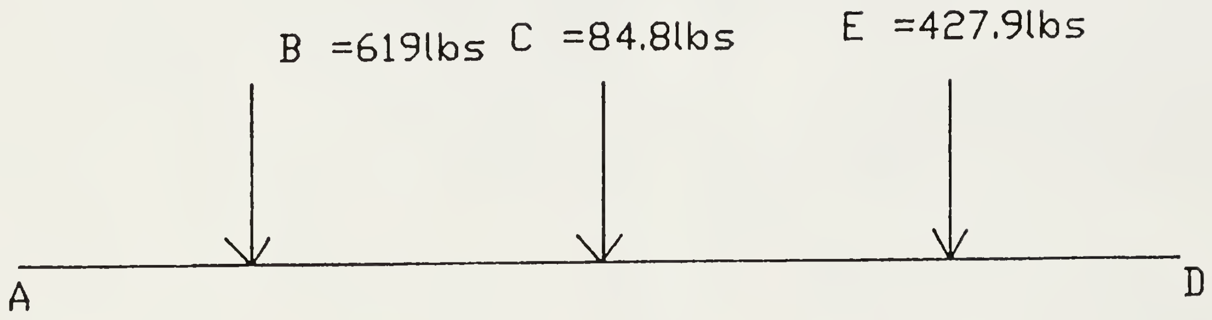



Figure 6.6(b) Vertical Load Diagram for single shaft of Fig.6.3

(iv)The distances of horizontal loads and vertical loads on the shaft from a fixed reference.

The above selection of design variables can be specified in the program through the ASSIGN routine using the statements given below.

```
SPOS(1)=B(1) ...cutting speed for turning shaft surface
SPIB(1)=B(2) ...cutting speed for turning bearing inner surface
SPOB(1)=B(3) ...cutting speed for turning bearing outer surface
SPIH(1)=B(4) ...cutting speed for turning housing hole
FDOS(1)=B(5) ...cutting feed rate for shaft surface
FDIB(1)=B(6) ...cutting feed rate for bearing inner surface
FDOB(1)=B(7) ...cutting feed rate for bearing outer surface
FDIH(1)=B(8) ...cutting feed rate for housing hole
UTOLOS(1)=B(9)  ...Upper tolerance on shaft outer diameter
LTOLOS(1)=B(10) ...Lower tolerance on shaft outer diameter
UTOLIB(1)=B(11) ...Upper tolerance on bearing inner diameter
LTOLIB(1)=B(12) ...Lower tolerance on bearing inner diameter
UTOLOB(1)=B(13) ...Upper tolerance on bearing outer diameter
LTOLOB(1)=B(14) ...Lower tolerance on bearing outer diameter
UTOLIH(1)=B(15) ...Upper tolerance on housing hole diameter
LTOLIH(1)=B(16) ...Lower tolerance on housing hole diameter
SLEN(1) = 50.   ...Length of shaft ( constant )
BLEN(1) = B(17) ...Length of bearing considered as a design variable
HLEN(1) = BLEN(1) ... Housing length being turned is made equal to
                      bearing length being turned
DIAOS(1)=B(18)  ...   Diameter of shaft surface
```

DIAIB(1)=DAIOS(1) .. Inner diameter of bearing set equal to the diameter of shaft surface

DIAOB(1)=B(19) ... Outer diameter of bearing

DIAIH(1)=DIAOB(1).. Diameter of housing hole set equal to outer diameter of bearing

AHDIS(1,1)=B(20) ... Distance of horizontal load1 from the fixed reference end

AHDIS(1,2)=B(21) ... Distance of H load2 from the fixed reference end

AHDIS(1,3)=B(22) ... Distance of H load3 from the fixed reference end

AVDIS(1,1)=AHDIS(1,1) .. Distance of V load1 should be the same as distance of H load1 from the fixed end

AVDIS(1,2)=AHDIS(1,2) .. Distance of V load2 should be same as distance of H load2 from the fixed end

AVDIS(1,3)=AHDIS(1,3) .. Distance of V load3 should be the same as distance of H load3 from the fixed end

The initial guess and the optimal values of the design vector are as shown in Table 6.4 .

RESULTS FOR EXAMPLE 6.3

| DESIGN VARIABLE | INITIAL GUESS | FINAL VALUE |
|---|---|---|
| SPOS(1) | 100.0 sfpm | 150.94 sfpm |
| SPIB(1) | 100.0 sfpm | 291.04 sfpm |
| SPOB(1) | 100.0 sfpm | 291.04 sfpm |
| SPIH(1) | 100.0 sfpm | 150.94 sfpm |
| FDOS(1) | 0.03 ipr | 0.0272 ipr |
| FDIB(1) | 0.04 ipr | 0.0153 ipr |
| FDOB(1) | 0.04 ipr | 0.0136 ipr |
| FDIH(1) | 0.03 ipr | 0.0290 ipr |
| UTOLOS(1) | -0.003 in | -0.00220 in |
| LTOLOS(1) | -0.0045 in | -0.00495 in |
| UTOLIB(1) | 0.005 in | 0.00505 in |
| LTOLIB(1) | 0.0035 in | 0.00350 in |
| UTOLOB(1) | -0.0035 in | -0.00100 in |
| LTOLOB(1) | -0.0045 in | -0.00255 in |
| UTOLIH(1) | 0.005 in | 0.00315 in |
| LTOLIH(1) | 0.003 in | 0.00160 in |
| BLEN(1) | 5.00 in | 5.00 in |
| DIAOS(1) | 2.5 in | 2.5591 in |
| DIAOB(1) | 3.25 in | 3.1989 in |

Table 6.4

| | | | | |
|---|---|---|---|---|
| AHDIS(1,1) | 12.00 | in | 10.05 | in |
| AHDIS(1,2) | 30.00 | in | 31.95 | in |
| AVDIS(1,3) | 42.00 | in | 43.95 | in |

---

| | | |
|---|---|---|
| Initial value of Pseudo Objective Function | = | 86.7623 |
| Final value of Pseudo Objective Function | = | 1.045 |
| Initial value of true cost | = | 3.76 $ |
| Final value of true cost | = | 1.045 $ |
| Sum of Initial Constraint Violations | = | 0.83002 E-2 |
| Sum of Final Constraint Violations | = | 0.4166 E-8 |

---

Table 6.4 (CONTD.)

## 6.4 Design and manufacture of two shaft transmission assembly :

A transmission shaft assembly with two shafts needs to be designed as shown in Fig. 6.7 . The various components of the assembly (the shaft, bearings and the housing) are to be manufactured by the turning process. The optimal set of design variables (machining parameters and component design parameters) needs to be found for the case of minimum cost of manufacture using the optimal design code described in earlier chapters. The following information gives the requirements and the specifications. The figures 6.7 and 6.8 show the details of this problem.

Pulley B (24 in. diameter) receives 40 hp at 360 rpm from below at an angle of 45 degrees as shown in Fig. 6.8 . The 8 inch gear D and a 12 inch gear C are fixed on the first shaft. Two gears E (12" dia.) and F(18" dia.) are mounted on the second shaft and a dog clutch engages one of the two gears to the shaft based upon the speed desired and the other one just rotates freely without being engaged to the shaft (i.e. without delivering any torque). It is required to find the optimal design of the shaft assembly by calculating the set of variables like the diameters of the components being used, the optimal loading distribution on the shaft and the bearing design parameters. Also, the optimal machining parameters are to be determined in order to obtain the design corresponding to the least possible manufacturing cost. The following are the manufacturing requirements desired :

Surface finish on outer surface of shafts : 50.0 mi.in. (rms)

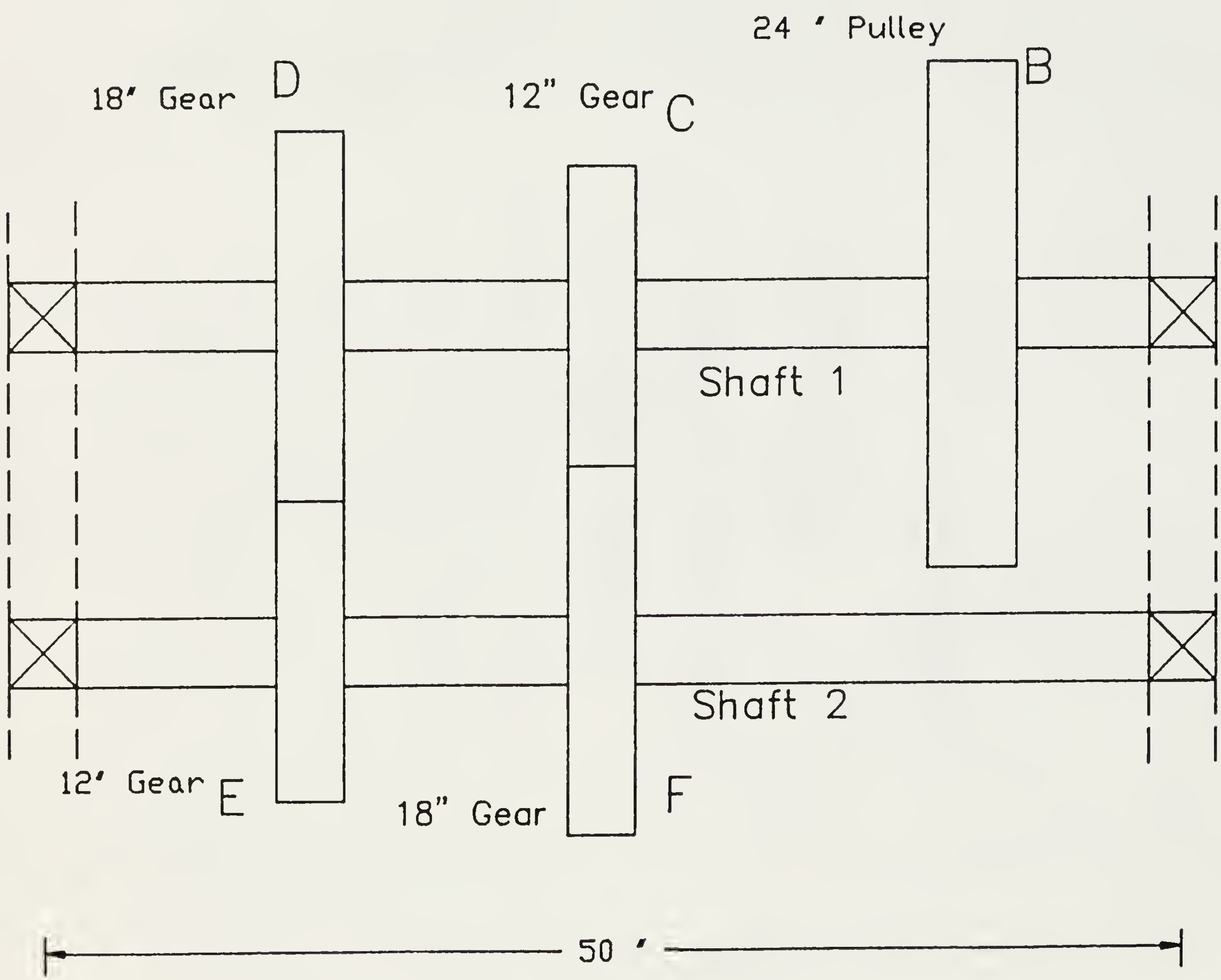



Figure 6.7 Double shaft assembly with transmission loads

Surface finish on inner surface of bearings : 50.0 mi.in. (rms)

Surface finish on outer surface of bearings : 50.0 mi.in. (rms)

Surface finish on inner surface of housing : 50.0 mi.in. (rms)

Maximum clearance between shaft outer surface and inner surface of bearing : 0.010 in.

Minimum clearance between shaft outer surface and inner surface of bearing : 0.0057 in.

Maximum clearance between bearing outer surface and housing hole : 0.0057 in.

Minimum clearance between bearing outer surface and housing hole : 0.002 in.

Minimum gap between tolerance values : 0.00155 in.

Nominal feeds for the turning of components : 0.04 ipr

Maximum tool nose radius : 0.6 in

Using the analysis methods discussed in the previous example, the torques at the various loads can be obtained from the free body diagrams as shown in Fig. 6.8 . The values of these torques are :

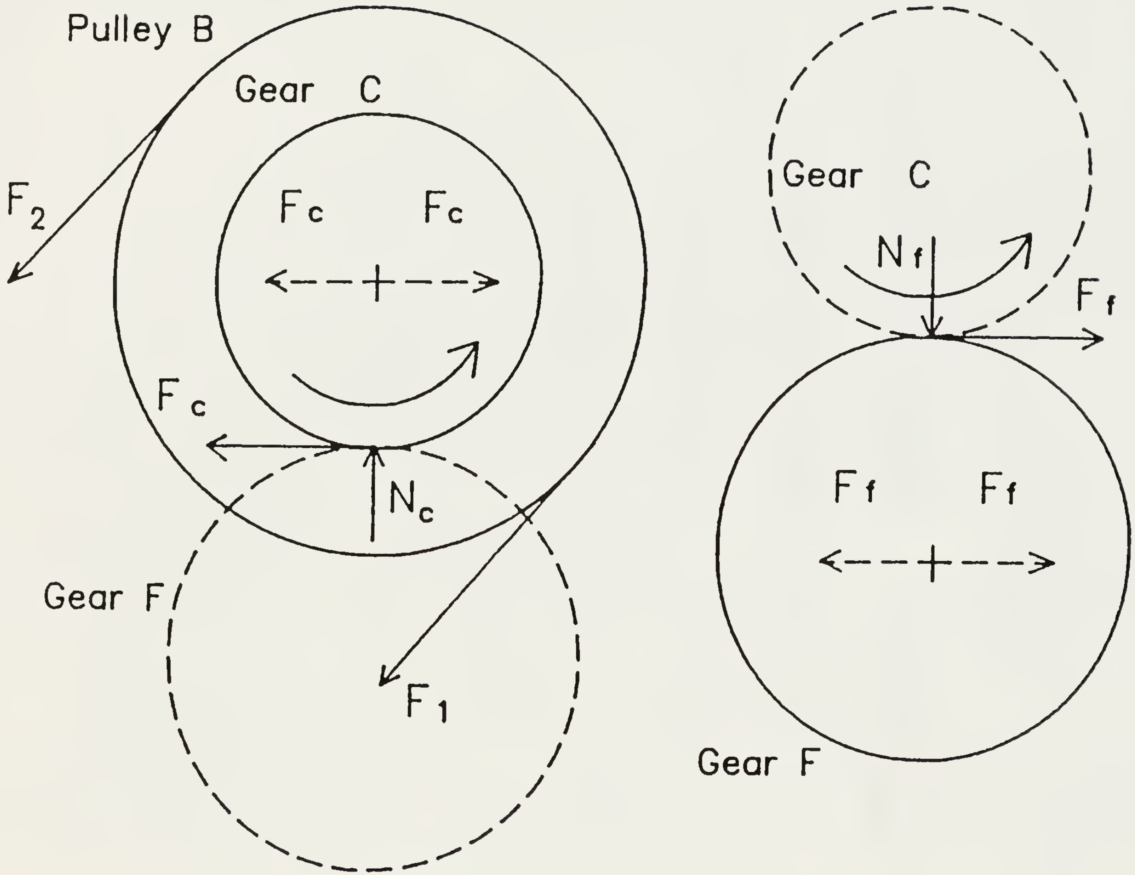
Torque at B = 63000(40)/360 = 7000.0 lb.-in.

If F is engaged to the shaft and E is moving freely,

$$T_c * \omega_c = T_f * \omega_f \qquad \text{(for losses} \approx 0\text{ )}$$

If $N_c$ and $N_f$ are the RPMs of the two shafts, this can be written as

$$(T_c * 2\pi\ N_c)/60 = (T_f * 2\Pi\ N_f)/60$$




FREEBODY DIAGRAMS FOR SHAFT1 & SHAFT2

Figure 6.8 Force diagrams for the double shaft assembly of Fig 6.7

i.e., $T_f = T_c(N_c/N_f)$

The ratio $N_c/N_f$ is called the speed ratio and is given by the following relations :

$$\text{Speed ratio} = N_c/N_f = \omega_c/\omega_f = D_f/D_c = T_f/T_c$$

where $\omega$'s represent the angular speeds,

D's represent the pitch diameters

and T's represent the number of gear teeth.

Therefore the speed ratio becomes 18/12 = 3/2

and

$$T_f = T_c(3/2) = 7000(3/2) = 10500 \text{ lb-in}$$

If gear E is engaged with gear D ,

$$T_e * \omega_e = T_d * \omega_d$$

and

$$T_e = T_d * (\text{speed ratio})$$

where speed ratio is given by $D_e/D_d$ = 12/18 =2/3

Similarly,

$$T_c = T_d * (2/3)$$

$$= 7000(2/3) = 4666.666 \text{ lb. in}$$

The torque being experienced by the upper shaft is the input torque. The second shaft will experience a torque that is 2/3 or 3/2 times the value of input torque based upon the gear (E or F) being engaged to the shaft. For the design of the shaft the worst case had to be considered. Hence the design is made for the case of the higher torque load i.e. $T_f$ =(10500in.-lb).

Thus the maximum torques for the two shafts are

$T_1$=7000 lb-in and $T_2$=10500 lb-in

The bending force produced by the belt around pulley is given by

$F_b=2(F_1-F_2)=2(T_b)/r_b=2(7000)/12 = 1166.66$ lb

For the gears, the driving forces are computed as though the contact is always on the pitch circle :

$F_e=T_e/r_e$ =7000/6 =1166.66 lb ($r_e$ being the pitch circle radius )

As shown in Fig. 6.5 , F acts tangential to the pitch circle and normal to the tooth. A separating force N comes into play because of F, and depends on the pressure angle $\phi$ ; thus, the separating force $N_c$ is given by

$$N_c = F_c \tan\phi$$

Similarly, for the gear F which is in mesh with gear C

$F_f$=7000/6=1166.66 lb

$N_f=N_c$=424.632 (the direction is opposite to that of $N_c$)

The gear E is not simultaneously engaged to the shaft hence no power or torque transmission takes place at E.

Now the forces are resolved into two planes H and V according to Fig. 6.8 as follows :

$F_c$=1166.66 lb

$F_b$=1166.66 lb

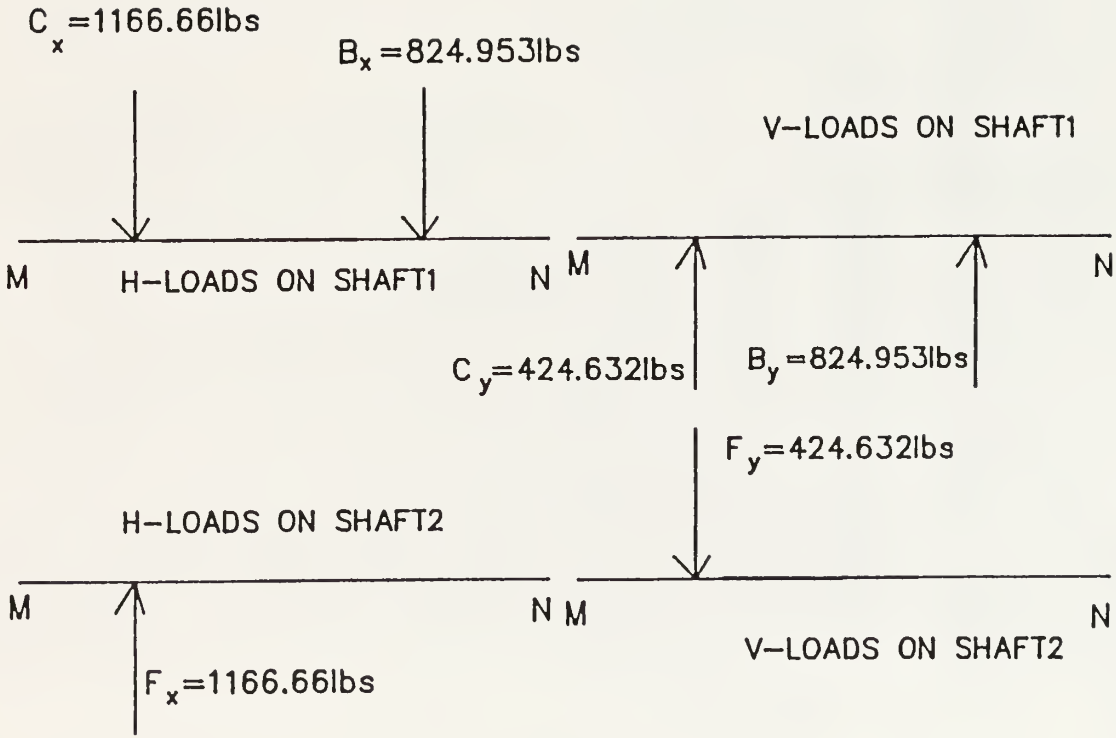



Figure 6.9 Horizontal and Vertical Loads acting on the shafts of Fig.6.7

Resolving the forces on shaft 1, the horizontal components are

$$C_x = F_c = 1166.66 \text{ lb}$$

$$B_x = F_b \cos 45 = 1166.66 \cos 45 = 824.953 \text{ lb}$$

Taking moments about M,

$$C_x(22) + B_x(32) - N_x(50) = 0$$

$N_x = +1041$ (positive sign indicates that the sense of $N_x$ is correct)

Now considering the second shaft as shown in Fig. 6.8

$$F_x = F_c = 1166.66 \text{ lb}$$

Similarly for the vertical direction

$$C_y = N_c = 424.632 \text{ lb}$$

$$B_y = F_b \cos 45 = 824.953 \text{ lb} \quad \text{(shaft 1)}$$

For the load F on second shaft

$$F_y = N_f = 424.632 \text{ lb} \qquad \text{(shaft 2)}$$

The diagrams for the shaft loads are as shown in Fig.6.9 . Thus the horizontal and vertical loads for this case are :

HPLOD(1,1)= 1166.66 lbs

HPLOD(1,2)= 824.953 lbs

HPLOD(2,1)=-1166.66 lbs

VPLOD(1,1)= 424.632 lbs

VPLOD(2,1)= 424.632 lbs

The results of this problem are shown in Table 6.5 .

RESULTS FOR EXAMPLE 6.4

| DESIGN VARIABLE | INITIAL GUESS | FINAL VALUE |
|---|---|---|
| SPOS(1) | 100.0 sfpm | 239.80 sfpm |
| SPIB(1) | 100.0 sfpm | 291.04 sfpm |
| SPOB(1) | 100.0 sfpm | 291.04 sfpm |
| SPIH(1) | 100.0 sfpm | 254.75 sfpm |
| FDOS(1) | 0.03 ipr | 0.0232 ipr |
| FDIB(1) | 0.04 ipr | 0.0162 ipr |
| FDOB(1) | 0.04 ipr | 0.0140 ipr |
| FDIH(1) | 0.03 ipr | 0.0234 ipr |
| UTOLOS(1) | -0.003 in | -0.00210 in |
| LTOLOS(1) | -0.0045 in | -0.00481 in |
| UTOLIB(1) | 0.005 in | 0.00519 in |
| LTOLIB(1) | 0.0035 in | 0.00359 in |
| UTOLOB(1) | -0.0035 in | -0.00100 in |
| LTOLOB(1) | -0.0045 in | -0.00255 in |
| UTOLIH(1) | 0.005 in | 0.00310 in |
| LTOLIH(1) | 0.003 in | 0.00100 in |
| SPOS(2) | 100.0 sfpm | 254.75 sfpm |
| SPIB(2) | 100.0 sfpm | 291.04 sfpm |
| SPOB(2) | 100.0 sfpm | 291.04 sfpm |

Table 6.5

| | | |
|---|---|---|
| SPIH(2) | 100.0 sfpm | 254.75 sfpm |
| FDOS(2) | 0.03 ipr | 0.0234 ipr |
| FDIB(2) | 0.04 ipr | 0.0168 ipr |
| FDOB(2) | 0.04 ipr | 0.0150 ipr |
| FDIH(2) | 0.03 ipr | 0.0234 ipr |
| UTOLOS(2) | -0.003 in | -0.00220 in |
| LTOLOS(2) | -0.0045 in | -0.00490 in |
| UTOLIB(2) | 0.005 in | 0.00505 in |
| LTOLIB(2) | 0.0035 in | 0.0035 in |
| UTOLOB(2) | -0.0035 in | -0.00100 in |
| LTOLOB(2) | -0.0045 in | -0.00255 in |
| UTOLIH(2) | 0.005 in | 0.00310 in |
| LTOLIH(2) | 0.003 in | 0.00100 in |
| BLEN(1) | 5.25 in | 5.00000 in |
| DIAOS(1) | 2.5 in | 2.435 in |
| DIAOB(1) | 3.25 in | 3.044 in |
| BLEN(2) | 4.25 in | 5.00000 in |
| DIAOS(2) | 2.00 in | 2.127 in |
| DIAOB(2) | 2.75 in | 2.659 in |
| AHDIS(1,1) | 12.00 in | 7.000 in |
| AHDIS(2,1) | 33.00 in | 42.200 in |

Table 6.5 (CONTD.)

| | | |
|---|---|---|
| Initial value of Pseudo Objective Function | = | 36459421.67 |
| Final value of Pseudo Objective Function | = | 11.091 |
| Initial value of true cost | = | 2.088 $ |
| Final value of true cost | = | 11.091 $ |
| Sum of initial constraint violations | = | 3645.941958 |
| Sum of final constraint violations | = | 0.1042E-8 |

Table 6.5 (CONTD.)

## 6.5 Design and Manufacture of Three shaft Assembly :

A transmission shaft assembly with three shafts needs to be designed as shown in the Fig. 6.10 . The various components of the assembly ( the shafts, bearings and the housing ) are to be manufactured by the turning process. The optimal set of design variables (machining parameters and component design parameters) needs to be found for the case of minimum cost of manufacture using the optimal design code described in the earlier chapters. The following information gives the requirements and the specifications. Fig. 6.10 shows the details of this problem.

Pulley B (24 in. diameter) receives 30 hp at 360 rpm from above at an angle of 45 degrees as in the previous problem. The 12 inch gear C delivers the power to 18 inch gear D. There are two more gears E (12" dia.) and G(18" dia.) mounted on the second shaft. On shaft 3 gears F and H rotate freely and one of the two can be engaged to the shaft by a moveable dog clutch, thus allowing two possible speeds for the shaft. It is required to find the optimal design of the shaft assembly, i.e. optimal values must be found for the diameters of the components being used, the loading distribution on the shaft and the bearing design parameters. Also the optimal machining parameters are to be determined in order to obtain the component design for the least possible manufacturing cost. The following are the manufacturing requirements desired :

Surface finish on outer surface of shafts : 50.0 mi.in. (rms)

Surface finish on inner surface of bearing : 80.0 mi.in. (rms)

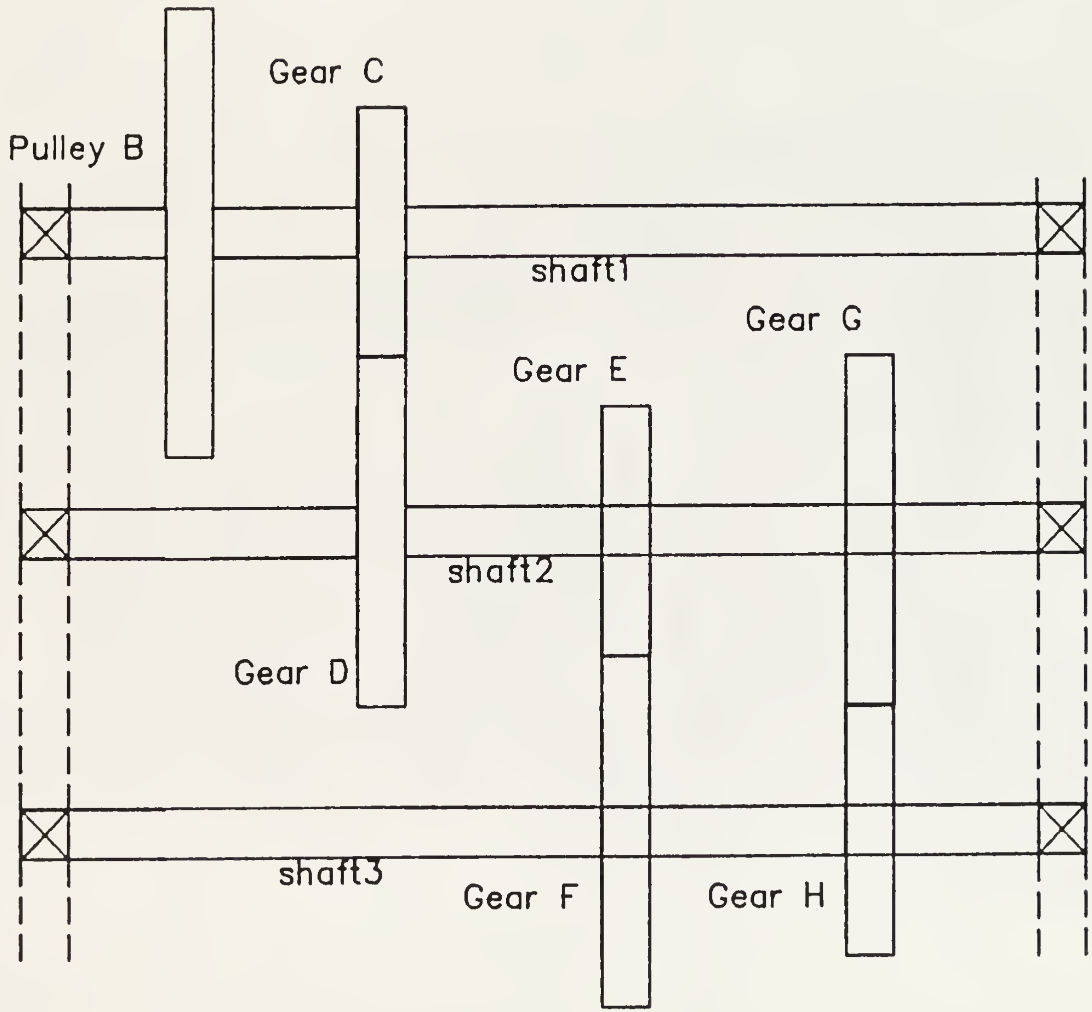



Figure 6.10 Three shaft assembly with transmission loads

Surface finish on outer surface of bearing : 59.0 mi.in. (rms)

Surface finish on inner surface of housing holes : 89.0 mi.in. (rms)

Maximum clearance between shaft outer surface and inner surface of bearing : 0.010 in.

Minimum clearance between shaft outer surface and inner surface of bearing : 0.0057 in.

Maximum clearance between bearing outer surface and housing hole : 0.0057 in.

Minimum clearance between bearing outer surface and housing hole : 0.002 in.

Minimum gap between upper & lower tolerances : 0.00155 in.

Nominal feeds for the turning of components : 0.04 ipr

Maximum tool nose radius : 0.6 in

Maximum Deflection of any shaft : 0.0062 in

Maximum Deviation of any shaft : 0.0275 in

Maximum Deviation of shaft1 & shaft2 from mean position : 0.05 in

Using the same procedure as illustrated in Examples 6.3 and 6.4, the loading can be resolved into the following horizontal and vertical systems as shown in Figs. 6.11 and 6.12 respectively .

HPLOD(1,1)=$B_x$=618.7184 lb.

HPLOD(1,2)=$C_x$=875 lb.

HPLOD(2,1)=$E_x$=-1312.5 lb.

HPLOD(2,2)=$D_x$=-875 lb.

HPLOD(3,1)=$F_x$=1312.51 lb.

VPLOD(1,1)=$B_y$=618.7184 lb.

VPLOD(1,2)=$C_y$=318.474 lb.

VPLOD(2,1)=$E_y$=477.71 lb.

VPLOD(2,2)=$D_y$=-318.474 lb.

VPLOD(3,1)=$F_y$=-477.71 lb.

Regarding the load distribution the loads on the first shaft can be placed first for convenience. However, the loads on the other shafts are dependent on the load distribution on the first shaft since the loads are meshing gears; hence the following relationships have to be incorporated into the ASSIGN routine.

AHDIS(1,1) ................ design variable

AHDIS(1,2) ................ design variable

AHDIS(2,2)=$D_x$=AHDIS(1,2) ... Hload2 on shaft1 and Hload2 on shaft2 act at the same distance from fixed reference end.

AHDIS(2,1) .............. this is also a design variable as it is not dependent on the loads on shaft1

AHDIS(3,1)=AHDIS(2,1)...... load1 on shaft3 is at the same location as load1 on shaft2.

Also the horizontal loads and the vertical loads act at the same point

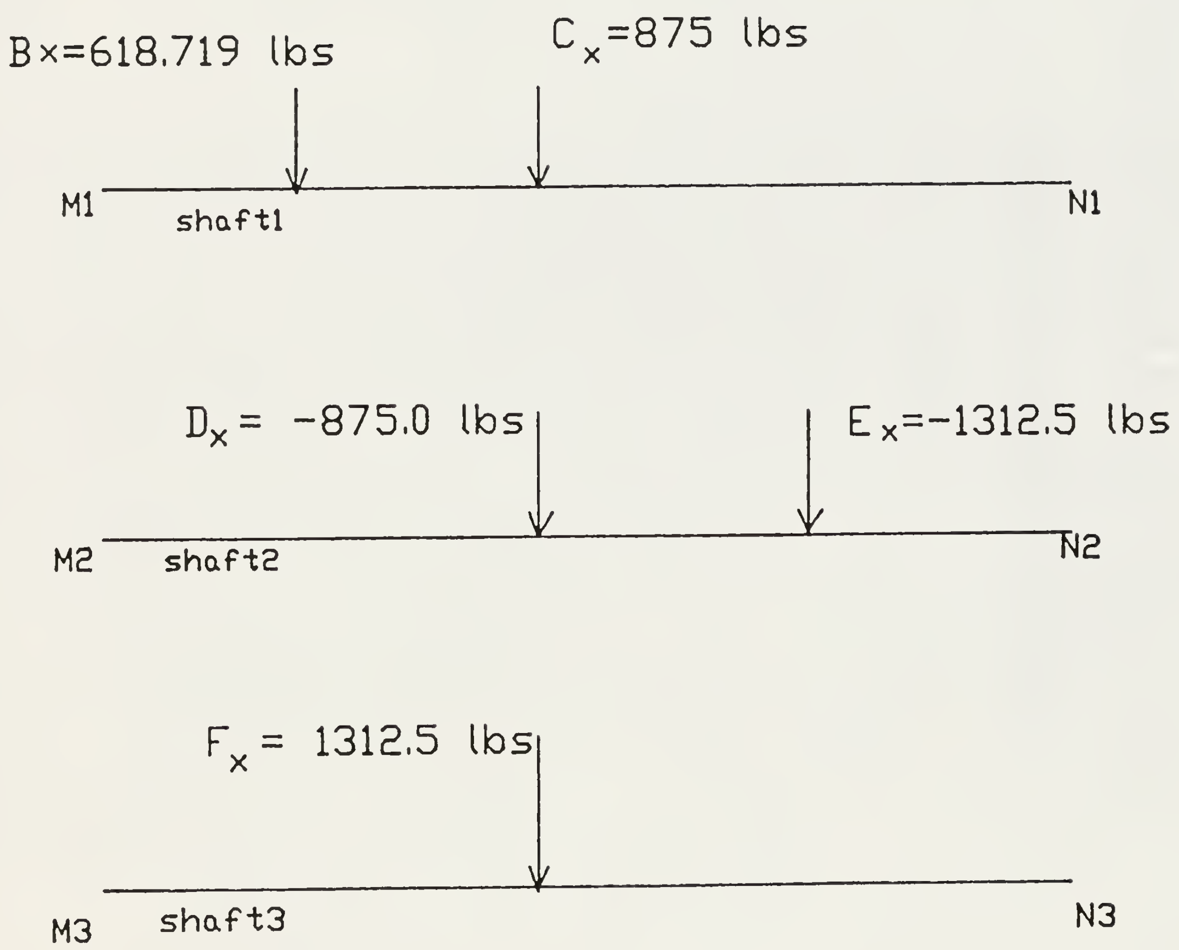



Figure 6.11 Horizontal loads acting on the three shafts of the assembly of Fig.6.10

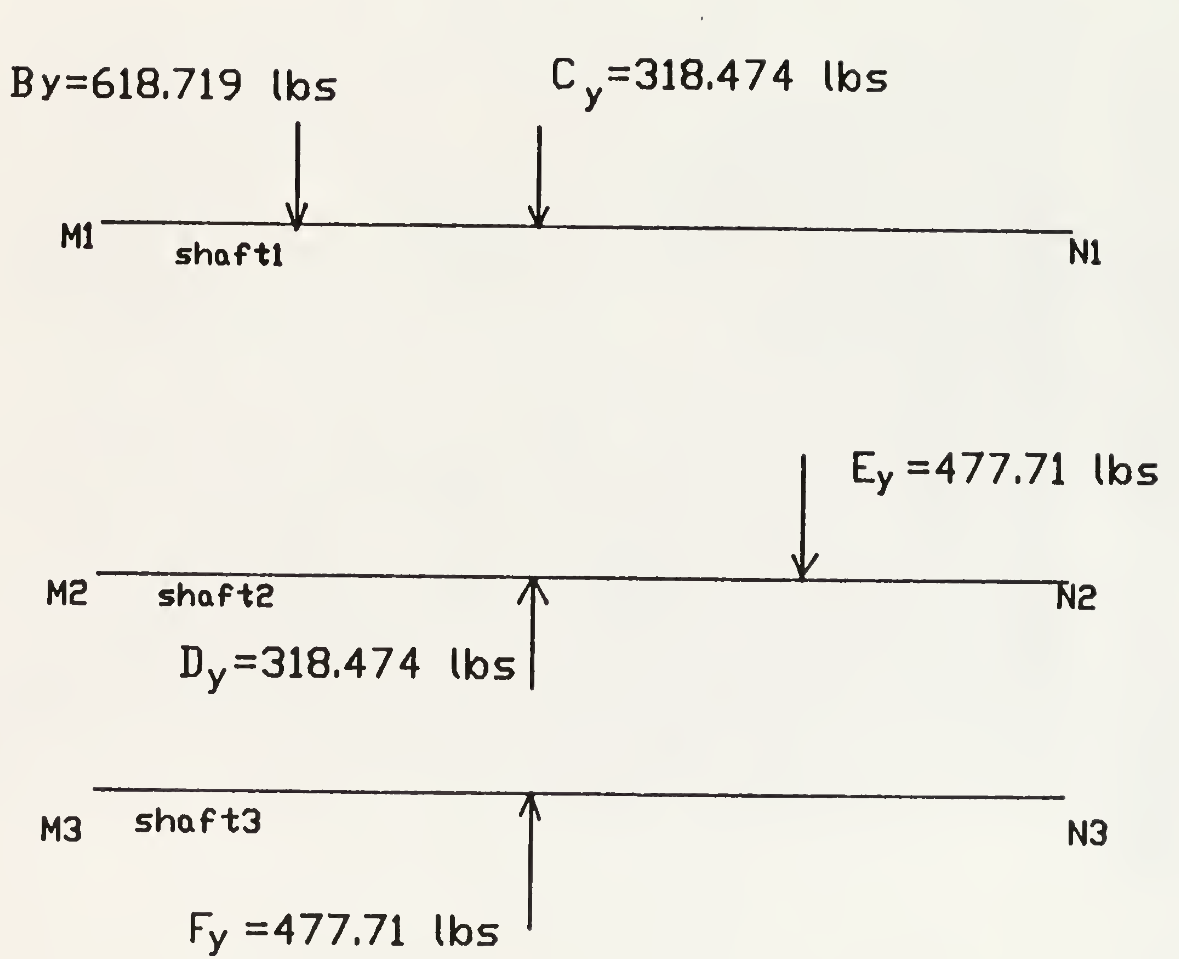



Figure 6.12 Vertical loads acting on the three shafts of the assembly of Fig.6.10

as they are derived from the same loads. Thus the following statements are needed in ASSIGN routine :

```
AVDIS(1,1) = AHDIS(1,1)
AVDIS(1,2) = AHDIS(1,2)
AVDIS(2,2) = AHDIS(2,2)
AVDIS(2,1) = AHDIS(2,1)
AVDIS(3,1) = AHDIS(2,1)
```

The results obtained in this example are presented in Table 6.6 .

RESULTS FOR EXAMPLE 6.5

| DESIGN VARIABLE | INITIAL GUESS | FINAL VALUE |
| --- | --- | --- |
| SPOS(1) | 100.0 sfpm | 239.81 sfpm |
| SPIB(1) | 100.0 sfpm | 291.04 sfpm |
| SPOB(1) | 100.0 sfpm | 291.04 sfpm |
| SPIH(1) | 100.0 sfpm | 254.75 sfpm |
| FDOS(1) | 0.03 ipr | 0.0232 ipr |
| FDIB(1) | 0.04 ipr | 0.0160 ipr |
| FDOB(1) | 0.04 ipr | 0.0130 ipr |
| FDIH(1) | 0.03 ipr | 0.0234 ipr |
| UTOLOS(1) | -0.003 in | -0.00260 in |
| LTOLOS(1) | -0.0045 in | -0.00510 in |
| UTOLIB(1) | 0.005 in | 0.00485 in |
| LTOLIB(1) | 0.0035 in | 0.00315 in |
| UTOLOB(1) | -0.0035 in | -0.00120 in |
| LTOLOB(1) | -0.0045 in | -0.00285 in |
| UTOLIH(1) | 0.005 in | 0.00255 in |
| LTOLIH(1) | 0.003 in | 0.00100 in |
| SPOS(2) | 100.0 sfpm | 224.73 sfpm |
| SPIB(2) | 100.0 sfpm | 291.04 sfpm |

Table 6.6

| | | |
|---|---|---|
| SPOB(2) | 100.0 sfpm | 291.03 sfpm |
| SPIH(2) | 100.0 sfpm | 254.75 sfpm |
| FDOS(2) | 0.03 ipr | 0.0229 ipr |
| FDIB(2) | 0.04 ipr | 0.0157 ipr |
| FDOB(2) | 0.04 ipr | 0.0126 ipr |
| FDIH(2) | 0.03 ipr | 0.0234 ipr |
| UTOLOS(2) | -0.003 in | -0.00270 in |
| LTOLOS(2) | -0.0045 in | -0.00520 in |
| UTOLIB(2) | 0.005 in | 0.00475 in |
| LTOLIB(2) | 0.0035 in | 0.00305 in |
| UTOLOB(2) | -0.0035 in | -0.00120 in |
| LTOLOB(2) | -0.0045 in | -0.00285 in |
| UTOLIH(2) | 0.005 in | 0.00255 in |
| LTOLIH(2) | 0.003 in | 0.00100 in |
| SPOS(3) | 100.0 sfpm | 254.75 sfpm |
| SPIB(3) | 100.0 sfpm | 291.04 sfpm |
| SPOB(3) | 100.0 sfpm | 291.04 sfpm |
| SPIH(3) | 100.0 sfpm | 254.75 sfpm |
| FDOS(3) | 0.03 ipr | 0.0234 ipr |
| FDIB(3) | 0.04 ipr | 0.0185 ipr |
| FDOB(3) | 0.04 ipr | 0.0138 ipr |
| FDIH(3) | 0.03 ipr | 0.0234 ipr |
| UTOLOS(3) | -0.003 in | -0.00265 in |

Table 6.6 (CONTD.)

| | | |
|---|---|---|
| LTOLOS(3) | -0.0045 in | -0.00515 in |
| UTOLIB(3) | 0.005 in | 0.00485 in |
| LTOLIB(3) | 0.0035 in | 0.00305 in |
| UTOLOB(3) | -0.0035 in | -0.00120 in |
| LTOLOB(3) | -0.0045 in | -0.00285 in |
| UTOLIH(3) | 0.005 in | 0.00255 in |
| LTOLIH(3) | 0.003 in | 0.00100 in |
| BLEN(1) | 5.25 in | 5.000 in |
| DIAOS(1) | 3.2 in | 2.763 in |
| DIAOB(1) | 3.7 in | 3.454 in |
| BLEN(2) | 4.25 in | 5.000 in |
| DIAOS(2) | 3.25 in | 2.940 in |
| DIAOB(2) | 3.90 in | 3.675 in |
| BLEN(3) | 4.25 in | 5.000 in |
| DIAOS(3) | 3.10 in | 2.364 in |
| DIAOB(3) | 3.50 in | 3.100 in |
| AHDIS(1,1) | 14.00 in | 8.540 in |
| AHDIS(1,2) | 28.00 in | 31.00 in |
| AHDIS(1,3) | 37.00 in | 39.52 in |

Initial value of Pseudo Objective Function = 13872074.489

Final value of Pseudo Objective Function = 17.233

Table 6.6 (CONTD.)

| | | |
|---|---|---|
| Initial value of true cost | = | 4.206 $ |
| Final value of true cost | = | 17.233 $ |
| Sum of initial constraint violations | = | 1387.2070283 |
| Sum of final constraint violations | = | 0.000000000 |

Table 6.6(CONTD.)

Thus the results show consistent satisfaction of the constraints after optimization. It is evident from the results that the value of the pseudo-objective function decreases considerably during the process of optimization. The optimized true value is attained after the satisfaction of all constraints; the initial true cost may sometimes be lower than the final cost but this is because the initial design is infeasible whereas the final design is not. Hence these numerical examples makes it clear that the set of machining parameters and the design parameters corresponding to the minimum cost of manufacture can be found while ensuring the satisfaction of all the design and manufacturing constraints.

# CHAPTER VII

## CONCLUSION

The primary aim of the research work presented in this thesis was to explore means to select optimal machining parameters for the manufacture of transmission shaft assembly components at the least possible cost while ensuring that all manufacturing requirements are met. This method was then extended to include the selection of optimum design parameters in addition to the manufacturing parameters for simultaneously satisfying design and manufacture specifications.

A nonlinear mathematical programming approach was used to achieve the objective of minimum manufacturing cost. Approximating functions were fitted to existing tables relating the cutting parameters to the tolerance and surface finish of the finished work piece. Using these functions a generalised constrained optimization problem for minimizing the manufacturing cost was formulated. The key manufacturing parameters of feed, speed and upper and lower manufacturing tolerances on part diameters were included in the design vector. Various manufacturing specifications like surface finish, fit requirements of mating parts and tolerance requirements were ensured through the application of constraints. The integration of design and manufacture was achieved by extending the formulation of the generalised constrained optimization problem to include design and manufacturing considerations within a single optimization problem. Design parameters like the dimensions of components and the locations of loads on the shaft were were also included as design variables. The design constraints which were added included the limits on shear stress, maximum deflection, maximum deviation, etc. Further, the approach was enhanced to be capable of handling single and multiple shaft problems. The optimization procedure adopted for the solution of the optimization problem was an exterior penalty function method using a directed grid search for unconstrained minimization. The above solution method was implemented in a computer aided design code that can be used in two ways : firstly it can be used for the determination of optimum machining parameters for a fixed design ; secondly, it can also be used for the simultaneous optimization of design and manufacturing parameters. The user has complete control over the selection of design variables and can assign any set of parameters to desired fixed values. The code thus gives the user the flexibility to experiment with many different combinations of variables and parameter values. Two classes of problems were solved using the developed code. First, manufacturing problems with fixed design specifications were solved to determine optimal machining parameters for minimum cost of manufacture. In the second class of problems design parameters were also included in the design vector. Additional examples were solved after integrating the design constraints into the optimization problem. Also multiple shaft problems were successfully solved for both classes of problems. The results obtained attest to the feasibility and efficacy of the technique developed in this thesis.

## 7.2 Suggestions for future work :

There is a strong need to pursue extensive experimental work in order to derive more accurate relationships between surface finish, manufacturing tolerance, tool life, and the independent machining parameters like cutting speed, feed, depth of cut, tool nose radius, etc. If such relationships are used in the mathematical programming models, the accuracy of the results can be further improved. Also some more constraints can be added to the optimization model developed in this thesis in order to include the influence of tool geometry on the manufacturing cost.

More recent design theories for shaft design may be used instead of the ASME shaft design code. Also, in order to obtain greater computational efficiency, a derivative based optimization technique can be used to minimize the cost of manufacture. There is also potential for the use of multiobjective optimization techniques for maximizing production rate and minimizing manufacturing cost at the same time. The effect of the temperatures encountered during machining operations and the effects of coolants on the surface conditions of the work piece should also be considered in the optimization model. Also the computer integrated optimization technique used in this work may be extended to consider design and manufacture of other assemblies and components as well as other machining processes like boring, milling, broaching, grinding, etc. Finally, a very general artificial intelligence based CIM model can be developed after collecting substantial knowledge about the various machining and design processes and using this as a basis for building knowledge based systems which could help the machinist in taking faster and better decisions.

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# COMPUTER INTEGRATED MANUFACTURE OF TURNED SHAFT ASSEMBLIES

by

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B.E. , Osmania University (India) 1986

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AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the
requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering
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1988

ABSTRACT

The conventional practice followed in the manufacture and design of machined components is to select machining parameters based upon experience. The selection is often conservative and thereby uneconomical. Furthermore, manufacturing conditions are seldom taken into account at the design stage. This work is an attempt to develop an integrated approach to the optimal design and manufacture of turned assemblies in order to obtain simultaneous optimization of design and machining parameters. First, approximating functions were fitted to existing tables relating the cutting parameters to the tolerance and surface finish of the finished work piece. Using these functions, a generalised constrained optimization problem for minimizing the manufacturing cost was formulated. The design variables for this problem include the cutting speeds, feed rates and upper and lower manufacturing tolerances on the part diameters. The manufacturing requirements like surface finish, tolerances and fit requirements (clearance/interference) of mating parts are imposed through constraint functions. The formulation of the generalised constrained optimization problem was then extended to integrate design and manufacture into a single optimization problem. Design parameters like the dimensions of components, and the locations of loads on the shaft were also included as design variables. The design constraints which were added included the limits on shear stress, maximum deflection, maximum deviation, etc. The formulation is capable of handling single and multiple shaft problems. An exterior penalty function with a directed grid search for unconstrained minimization was used to solve the manufacturing optimization problem as well as the integrated design-manufacture optimization problem. A reliable computer program for the automatic formulation and solution of these problems was also developed. Several example problems were solved using this program. The results demonstrate the feasibility and efficacy of the methods developed in this thesis.