Trade Of Analysis For Helical Gear Reduction Units

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Abstract

Multi criteria optimization plays a very important role in product development. During The product concept phase, the designer decides a set of objectives that needs to be optimized. Every designer strives to achieve a utopian design that will optimize all the objectives. In real Life such a design is impossible to achieve. One of the reasons is the conflicting nature of Objectives that need to be optimized. The designer has to arrive at a compromised design that will satisfy the requirement to the greatest extent. Compromise involves tradeoffs between efficiency, cost, quality and other attributes.

The method presented here shows the implementation of a multi-criteria optimization technique. Various techniques that are used are presented and later one of them is illustrated using a gear train optimization problem. Numerous solutions are obtained by the analysis and they all are optimal solutions. To choose between the designs is one of the difficult tasks and it depends on the designer's judgment. There are available methods that provide valuable information about each design, and it helps the designer in making correct decision. The proposed study incorporates one such decision technique. The approach provides valuable insight in solving real life design problems using multi-criteria optimization technique. This method can be a part of an expert system built for gear-train design.

Index Terms: *mat lab, utopian design.*

1. Introduction:

1.1 Design Process

A design process is a process of determining structural properties which includes geometry and material that will satisfy all the functional requirement of a prototype. There are numerous processes available that help the decision-maker in deciding upon a good design that will fulfill most of the requirements. The basic steps in a design process. The starting step, Conceptualize the requirements usually consists of some imprecise statements of requirements. Research on the possibility of developing such a product is necessary, after which it is possible to restate the Objective. Constraints are formulated to bind the design process from generating infeasible designs. The Formulation and Solution of the problem is done using the objective and constraints and various alternative solutions are obtained. In the next step, all the possible solutions are analyzed and weighed against some reference to accept, reject or modify them. The preferred solution. An Engineering Drawing of the resulting solution is done. The next step is Prototyping followed by Production.

1.2 A Design Process

Conceptualize the requirements, Research, Objective, Constraints, Formulation and Solution, Analysis, Selection, Drawing, Proto-typing, Production.

The job of the designer is not simply to compute various parameters associated with the design. Decisions taken during the process affect the quality of the developed product. These decisions are complex in nature involving information that come from different sources and disciplines. In general, the decision to choose any particular design solution from the available set of solutions is based on more than one criteria or objective. Such problems are defined as multi-objective optimization problems. A designer has to use his judgment and experience to find the best solution for such problems.

2. Optimization

Optimization is a process of finding the best solution which is subjected to specific circumstances. This is an integral part of design process. During the design process, the decision-maker sets up an objective which is decided according to the functions to be performed by the product, e.g., an objective can be to maximize the heat flow through the pipes of a heat exchanger. This objective is then expressed in terms of some parameters, the results are evaluated against some criteria, and a solution is obtained that will achieve the desired objectives. In other words, optimization is a process of obtaining the design parameters that will maximize or minimizes the objectives set by the designer.

2.1 Types of Optimization

Most of the optimization problems are classified as follows:

A) According to the formulation
   1) Classical non-linear formulation
   2) Linear programming formulation
   3) Goal programming formulation

B) The basic classification includes
   1) Constrained versus unconstrained
   2) Linear versus non-linear
   3) Single objective versus multi-objective

C) Classification can also be made by the variables
   1) Continuous,2) Discrete,3) Mixed- integer.

3.0 Multi objective optimization:

3.1 Introduction.

Real engineering design problems are generally characterized by the presence of many, often conflicting and incommensurable objectives. This raises the issue about how different objectives should combine to yield a final solution. There is also the question on how to search for an optimal solution to the design problem. Thus, a multi-objective problem presents an inherent difficulty in the solution process. It is rarely possible to obtain a single optimal solution for such type of problems.

3.2 Different ways to perform multi-objective optimization Problems.

As mentioned earlier, real engineering design problems are usually characterized by the presence of many conflicting objectives that the design has to fulfill.

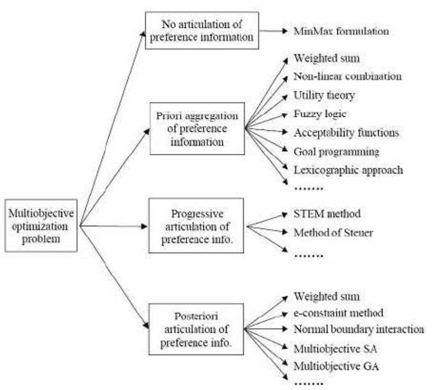


Fig-1: Multi objective optimization types.

A classification of some methods for multi-objective optimization.

3.2.1. Non-Linear Approaches.

Many methods can be classified as non-linear method for optimization. One of such method is Represented by the equation

$$\min \sum_{j=1}^{k}\left(\frac{fj(x)}{fjo}\right)^{p} \quad (1)$$

s.t. x ε s

In this method each objective functions are normalized and raised to exponent *p* . The exponent *p* expresses how much an improvement in f_i is worth and how much a poorer value penalizes the overall objective function.

3.2.2. Global Criterion Method.

In this method, the decision maker uses an approximate solution *f o* to formulate the mathematical model. He either uses an ideal solution *f o* if it is available or replaces it with the so called demand level *f d* that is specified by the decision maker. The objective function is written as

$$\min \sum_{i=1}^{N}\left(\frac{f0i - fi(x)}{f0i}\right)^{p} \quad (2)$$

s.t. $g_j(\mathbf{x}) \leq 0$, j=1,..., j

3.2.3. Goal Programming Method.

In this method, objectives are the goals with target values. The user then assigns weighting factors to rank the goals in order of importance. Finally a single objective function is written as the minimization of the deviations from the above stated goals. A "goal constraint" is slightly different than a "real constraint" in goal programming problems. A "goal constraint" is a constraint that the designer would like to be satisfied, but a slight deviation above or below this constraint is acceptable.

4.0 The gear train optimization problem:

4.1 Introduction

From the earlier chapter, we have observed that multi-objective optimization and decision theory play an important role in all the stages of design process. This theory and methodology is implemented in this work through a gear train transmission problem.

Gear train transmission units are a very important group of machine members and find variety of applications in today's world. Many automobile and aerospace applications use a multi-stage gear-box as the primary source of transmission. In this regard, the design of gear-box is of prime importance. All the members of gear-box must satisfy very rigorous technical requirements related to the reliability, efficiency and precise manufacturing of bearings and gears. The main parameters which are used to design a gearbox are the speed ratio and the transmitted torque.

4.2 Proposed Method

The work demonstrates the application of nonlinear multi-criteria optimization technique, with purpose to develop a methodology that can be included as a module into the gear design system. The work also studies the *tradeoff* between surface fatigue life and volume of a multi-stage helical gearbox. The study is limited to two and three stage gear reduction units although it can be applied to any multi-stage units. All the nomenclature used here is same as that found in.

4.3 The Formulation of Problem

In this work, a tradeoff between the geometrical volume and surface fatigue factor of a gear pair is analyzed. Proper attributes are selected to complete the design and perform optimization analysis. The attributes includes Objective, design variables and corresponding constraints. Design variables are identified that affect the design of the gearbox. The objective functions are chosen that are to be minimized. In this work, weight and surface fatigue life are the two objectives that are minimized. In general, these two are conflicting in nature. Constraints as well as the upper and lower bounds that are set for the design variable are decided to completely define the optimization problem.

4.4 Input Parameters

The input parameters were driving torque (T_{in}), speed ratio (e), normal pressure angle and design parameters. By varying the driving torque and speed ratio one at a time, Pareto optimal curves are generated. For analysis purpose, one of the parameter is held constant and the other is varied to generate different sets of Optimal Curves. The values considered are as follows.

Table 1: Input Parameters

Driving Torque (T_{in})	**Reduction Ratio (e)**
80	0.15
120	0.1
180	0.0667
270	0.05

4.5 Gear Design Parameters

Other design parameters and constants necessary for the formulation of constraint equations and Objective functions are provided in the following table.

Table 2: Gear Design Parameters

Description	**Symbol**	**Value**	**Units**
Bending Reliability Factor	k_r	0.814	None
Elastic Coefficient	C_p	2300	*psi*
Mean Stress Factor	k_{ms}	1.4	None
Mounting Factor	K_m	1.6	None
Overload Factor	K_o	1.0	None
Pressure Angle	-	30	degree
Shaft Length (N=3)	L_s	8.0	In.
Shaft Length (N=2)	L_s	4.0	In.
Surface Fatigue Strength	S_{fe}	190,000	Psi
Surface Reliability Factor (99%)	C_r	1.0	None
Torsional stress limit	max	25,000	Psi
Velocity factor	K_v	2.0	none

4.6 Design Variables.

The design variables are chosen in such a way that after the formulation of the problem the design intent is captured. All the design variables should be independent. During the design of gears, the most important parameters are diametral pitch and diameter of the gears. The design vector used in the three-stage gear train analysis is as follows.

$$X = (d_{p1}\ d_{g1}\ b_1\ P_{d1}\ P_{g1}\ T_{d1}\ T_{g1}\ m_{d1}\ m_{g1}\ i_1\ H_1\ \Psi_1\ d_{p2}\ d_{g2}\ b_2\ P_{d2}\ P_{g2}\ T_{d2}\ T_{g2}\ m_{d2}\ m_{g2}\ i_2\ H_2\ \Psi_2\ d_{p3}\ d_{g3}\ b_3\ P_{d3}\ P_{g3}\ T_{d3}\ T_{g3}\ m_{d3}\ m_{g3}\ i_3\ H_3\ \Psi_3)$$

Where

d_{p1} = Pinion diameter of the first gear set

d_{g1} = Gear diameter of the first gear set

b_1 = Face width of the first gear set

P_{d1} = Diametral Pitch of the first gear set

P_{g1}= Gear diameter pitch of the first gear set

T_{d1}= No. of teeth of the first gear set

T_{g1}= No. of teeth of the first gear set

m_{d1}= module of the pinion diameter of first gear set

m_{g1}= module of the gear diameter of first gear set

i_1 = gear ratio of first gear set

H_1 = Hardness of the first gear set

Ψ_1 = Helix angle of the first gear set

d_{p2} = Pinion diameter of the second gear set

d_{g2} = Gear diameter of the second gear set

b_2 = Face width of the second gear set

P_{d2} = Diametral Pitch of the second gear set

P_{g2}= Gear diameter pitch of the second gear set

T_{d2}= No. of teeth of the second gear set

T_{g2}= No. of teeth of the second gear set

m_{d2}= module of the pinion diameter of second gear set

m_{g2}= module of the gear diameter of second gear set

i_2 = gear ratio of second gear set

H_2 = hardness of the second gear set

ψ_2 = helix angle of the second gear set

d_{p3} = Pinion diameter of the third gear set

b_3 = Face width of the third gear set

P_{d3} = Diametral Pitch of the third gear set

P_{g3}= Gear diameter pitch of the third gear set

T_{d3}= No. of teeth of the third gear set

T_{g3}= No. of teeth of the third gear set

m_{d3}= module of the pinion diameter of third gear set

m_{g3}= module of the gear diameter of third gear set

H_3 = Hardness of the third gear set

Ψ_3 = Helix angle of the third gear set

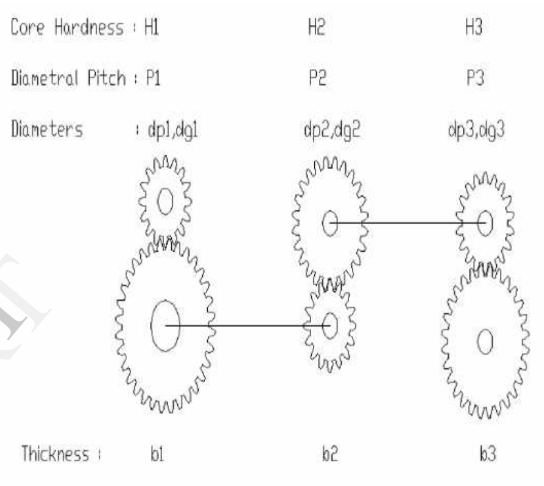


Figure-2: 3-stage Gear train.

Since the overall speed ratio for the analysis is decided, one of the gear diameters can be expressed in terms of other diameters using the speed ratio as follows.

$$d_{g3} = \left(\frac{d_{p1}\, d_{p2} d_{p3}}{d_{g1}\, d_{g2}\, e}\right) \qquad (3)$$

4.6.1 First gear set of upper bound

$\Psi1=14^{o}$

H1= 400

$d_{p1=}$20mm

$p_{d1=}$10mm

No. of teeth of pinion diameter of first gear set

$$T_{p1}= \frac{\pi\, d_{p1}}{P_{d1}} \qquad (4)$$

$T_{p1} = \frac{3.14 \times 20}{10}$

$T_{p1} = 6.28$

$T_{p1} = 7$

Module of the pinion diameter of the first gear set

$$m_{p1} = \frac{d_{p1}}{T_{p1}} \quad (5)$$

$m_{p1} = \frac{20}{7}$

$m_{p1} = 4mm$

$d_{g1} = 30mm$

$p_{g1} = 13mm$

No. of teeth of gear diameter of first gear set

$$T_{g1} = \frac{\pi \, d_{g1}}{p_{g1}} \quad (6)$$

$T_{g1} = \frac{3.14 \times 30}{13}$

$T_{g1} = 8$

Module of the gear diameter of the first gear set

$$m_{g1} = \frac{d_{g1}}{T_{g1}} \quad (7)$$

$m_{g1} = \frac{30}{8}$

$m_{g1} = 4mm$

Face width of first gear set

$b_1 = \pi \, m \cos\alpha$

$b_1 = 3.14 \times 4 \times \cos30$

$b_1 = 10$ mm

$$\text{Gear ratio (i)} = \frac{\text{gear diameter}}{\text{pinion diameter}} \quad (8)$$

$$\text{Gear ratio } (i_1) = \frac{d_{g1}}{d_{p1}} \quad (9)$$

$(i_1) = \frac{30}{20}$

$(i_1) = 1.5$

4.6.2 Second gear set of upper bound

$\Psi_2 = 20^o$

$H_2 = 400$

$d_{p2} = 25$ mm

$p_{d2} = 12$ mm

No. of teeth of pinion diameter of second gear set

$$T_{p2} = \frac{\pi \, d_{p2}}{p_{d2}} \quad (10)$$

$T_{p2} = \frac{3.14 \times 25}{12}$

$T_{p2} = 7$

Module of the pinion diameter of the second gear set

$$m_{p2} = \frac{d_{p2}}{T_{p2}} \quad (11)$$

$m_{p2} = \frac{25}{6}$

$m_{p2} = 5$ mm

$d_{g2} = 40$ mm

$p_{g2} = 14$ mm

No. of teeth of gear diameter of second gear set

$$T_{g2} = \frac{\pi \, d_{g2}}{p_{g2}} \quad (12)$$

$T_{g2} = \frac{3.14 \times 40}{14}$

$T_{g2} = 9$

Module of the gear diameter of the second gear set

$m_{g2} = \frac{d_{g2}}{T_{g2}}$ (13)

$m_{g2} = \frac{40}{9}$

$m_{g2} = 5$ mm

Face width of second gear set

$b_2 = \pi\ m\ \cos\alpha$ (14)

$b_2 = 3.14 \times 5 \times \cos 30$

$b_2 = 13$ mm

Gear ratio $(i_2) = \frac{d_{g2}}{dp2}$ (15)

$(i_2) = \frac{40}{25}$

$(i_2) = 1.6$

4.6.3 Third gear set of upper bound.

$\Psi_3 = 24^o$

$H_3 = 400$

$d_{p3} = 30$ mm

$p_{d3} = 14$ mm

No. of teeth of pinion diameter of third gear set

$T_{p3} = \frac{\pi\ d_{p3}}{p_{d3}}$ (16)

$T_{p3} = \frac{3.14 \times 30}{14}$

$T_{p3} = 8$

Module of the pinion diameter of the third gear set

$m_{p3} = \frac{d_{p3}}{T_{p3}}$ (17)

$m_{p3} = \frac{30}{8}$

$m_{p3} = 5$ mm

No. of teeth of gear diameter of third gear set

$T_{g3} = \frac{\pi\ d_{g3}}{p_{g3}}$ (18)

$T_{g3} = \frac{3.14 \times 45}{17}$

$T_{g3} = 10$

Module of the gear diameter of the third gear set

$m_{g3} = \frac{d_{g3}}{T_{g3}}$ (19)

$m_{g3} = \frac{45}{17}$

$m_{g3} = 5$ mm

Face width of third gear set

$b_3 = \pi\ m\ \cos\alpha$ (20)

$b_3 = 3.14 \times 5 \times \cos 30$

$b_3 = 13$ mm

Gear ratio $(i_3) = \frac{d_{g3}}{dp3}$ (21)

$(i_3) = \frac{45}{30}$

$(i_3) = 1.5$

4.6.4 First gear set of lower bound

$\Psi 1 = 8^o$

H1= 200

$d_{p1} = 10$mm

$p_{d1} = 5$mm

No. of teeth of pinion diameter of first gear set

$T_{p1} = \frac{\pi\ d_{p1}}{p_{g3}}$ (22)

$T_{p3} = \frac{3.14 \times 10}{5}$

T_{p1}= 7

Module of the pinion diameter of the first gear set

$m_{d1} = \frac{d_{p1}}{T_{p1}}$ (23)

$m_{d1} = \frac{10}{7}$

m_{d1} = 2 mm

d_{g1}=10mm

p_{g1}= 5 mm

No. of teeth of gear diameter of first gear set

$T_{g1} = \frac{\pi d_{g1}}{p_{g1}}$ (24)

$T_{g1} = \frac{3.14 \times 10}{5}$

T_{g1} = 7

Module of the gear diameter of the first gear set

$m_{g1} = \frac{d_{g1}}{T_{g1}}$ (25)

$m_{g1} = \frac{10}{5}$

m_{g1} = 2 mm

Face width of first gear set

$b_1 = \pi\, m \cos\alpha$ (26)

$b_1 = 3.14 \times 2 \times \cos 30$

b_1= 6 mm

Gear ratio (i) = $\frac{gear\ diameter}{pinion\ diameter}$

Gear ratio (i_1) = $\frac{d_{g1}}{d_{p1}}$ (27)

(i_1) = $\frac{10}{10}$

(i_1) = 1

4.6.5 Second gear set of lower bound

Ψ_2=14º

H_2= 200

d_{p2}=15 mm

p_{d2}=7 mm

No. of teeth of pinion diameter of second gear set

$T_{p2} = \frac{\pi d_{p2}}{p_{d2}}$ (28)

$T_{p2} = \frac{3.14 \times 15}{7}$

T_{p2}= 8

Module of the pinion diameter of the second gear set

$m_{p2} = \frac{d_{p2}}{T_{p2}}$ (29)

$m_{p2} = \frac{15}{7}$

m_{p2} = 3 mm

d_{g2}=20 mm

p_{g2}= 9 mm

No. of teeth of gear diameter of second gear set

$T_{g2} = \frac{\pi d_{g2}}{p_{g1}}$ (30)

$T_{g2} = \frac{3.14 \times 20}{9}$

T_{g2}= 8

Module of the gear diameter of the second gear set

$m_{g2} = \frac{d_{g2}}{T_{g2}}$ (31)

$m_{g2} = \frac{20}{9}$

m_{g2} = 3 mm

Face width of second gear set

$b_2 = \pi$ m cosα (32)

$b_2 = 3.14 \times 3 \times \cos 30$

$b_2 = 8$ mm

Gear ratio $(i_2) = \frac{d_{g2}}{dp2}$ (33)

$(i_2) = \frac{20}{15}$

$(i_2) = 1.3$

4.6.6 Third gear set of lower bound

$\Psi_3 = 18^o$

$H_3 = 200$

$d_{p3} = 20$ mm

$p_{d3} = 9$ mm

No. of teeth of pinion diameter of third gear set

$$T_{p3} = \frac{\pi\, d_{p3}}{p_{d3}} \quad (34)$$

$$T_{p3} = \frac{3.14 \times 20}{9}$$

$T_{p3} = 7$

Module of the pinion diameter of the third gear set

$$m_{p3} = \frac{d_{p3}}{T_{p3}} \quad (35)$$

$$m_{p3} = \frac{20}{7}$$

$m_{p3} = 4$ mm

No. of teeth of gear diameter of third gear set

$$T_{g3} = \frac{\pi\, d_{g3}}{p_{g3}} \quad (36)$$

$$T_{g3} = \frac{3.14 \times 25}{14}$$

$T_{g3} = 7$

Module of the gear diameter of the third gear set

$$m_{g3} = \frac{d_{g3}}{T_{g3}} \quad (37)$$

$$m_{g3} = \frac{25}{7}$$

$m_{g3} = 4$ mm

Face width of third gear set

$b_3 = \pi$ m cosα (38)

$b_3 = 3.14 \times 4 \times \cos 30$

$b_3 = 10$ mm

Gear ratio $(i_3) = \frac{d_{g3}}{dp3}$ (39)

$(i_3) = \frac{25}{20}$

$(i_3) = 1.2$

For a three stage design the bounds are
Upper bound:
x_{ub}=(2010,13,7,8,4,4,1.5,400,14,25,40,13,12,14,7,9,5,5,1.6,400,20,30,45,13, 14,17,8,10,5,5,1.5,400,24)

Lower Bound:
x_{lb}=(10,6,5,5,7,7,2,2,1,200,8,15,20,8,7,8,3,3,1.3,200,14,20,25, 10,9,14,7,7,4,4,1.2,200,18)

4.7 Constraint Equation Formulation

For any design problems there are two types of constraints: geometric and material constraints. These equations are needed to be expressed in terms of the design variables defined in the earlier section.

4.7.1 Material Constraints-Tooth Bending Fatigue Failure.

The gear tooth can be assumed as a cantilever with a force applied on the tip generated due to the driving torque. The modified Lewis equation is used to determine the stresses in the gear.

$$\sigma = \frac{F\,tp}{bj} K_v K_o\,(0.93K_m) \quad (40)$$

Where

Ft = Tangential gear force

J = Lewis geometry factor which is a function of helix angle, pressure angle and number of

Teeth of the pinion and the mating gear.

Kv = Velocity or dynamic factor. This factor indicates the severity of impact when successive

S=Gear pair gets engaged. It is function of pitch line velocity and manufacturing precision.

Ko = Overload Factor. It reflects the non-uniformity of driving and load torques.

Km = Mounting Factor. It determines the accuracy of alignment between the driving and driven Gears.

$$\frac{F_t F}{bj} K_v K_o (0.93K_m) \leq S'n \; C_s K_r K_{ms} \quad (41)$$

4.8 Geometric Constraints

4.8.1 Minimum Pinion Tooth Number

In general design, the minimum tooth number for a pinion is chosen.

$$N_p \geq 16 \quad (42)$$

This constraints for the three gear set is,

$$g_{15} = Pd_{p1} > 16 \quad (43)$$

$$g_{16} = Pd_{p2} > 16 \quad (44)$$

$$g_{17} = Pd_{p3} > 16 \quad (45)$$

4.9 Objective Function Formulation

Objective functions are the goals set be the decision maker to be achieved by fulfilling the constraint equations. In this analysis, we have chosen two objectives. The first objective is to minimize the overall volume (weight) of the gearbox. The second objective is to maximize the fatigue life.
The following section develops the necessary objective functions used for analysis

4.9.1 Tooth Surface Fatigue Failure

During traditional design, a particular fatigue life of the material is assumed a priori. But, this might result in incorrect design as the limiting parameters are either empirical or experimentally derived and the failure is not well-defined. Due to this reason, it will be appropriate to use the fatigue life as objective function, instead of a hard constraint. Gear teeth are subject to Hertz contact stresses which is given by

$$\sigma_H = C_p \sqrt{\frac{Ft\cos\psi}{bdI}} K_v K_o (0.93K_m) \quad (46)$$

$$\sigma_H = 2.3\sqrt{\frac{8 \times \cos 30}{4 \times 20 \times 0.20}} \times 2 \times 1(0.93 \times 1.6)$$

$$\sigma_H = 2.3\sqrt{\frac{8 \times 0.86}{4 \times 20 \times 0.20}} \times 2 \times 1(1.48)$$

$$\sigma_H = 2.3\sqrt{\frac{6.00}{16}} \times 2.96$$

Where

C_p = elastic coefficient of the material

d = diameter of the pinion.

I = Geometry Factor which is given by

$$I = \frac{1}{2}\sin\alpha\cos\alpha \quad (47)$$

$$I = \frac{1}{2}\sin 30\cos 30 \frac{40}{40+2.3}$$

$$I = \frac{1}{2} \times \frac{1}{2} \times 0.86(0.94)$$

$$I = \frac{1}{4} \times 0.80$$

$$I = 0.20$$

$$\sigma_H = 2.3\sqrt{\frac{6.88}{16}} \times 2.96$$

$$\sigma_H = 2.3\sqrt{1.27}$$

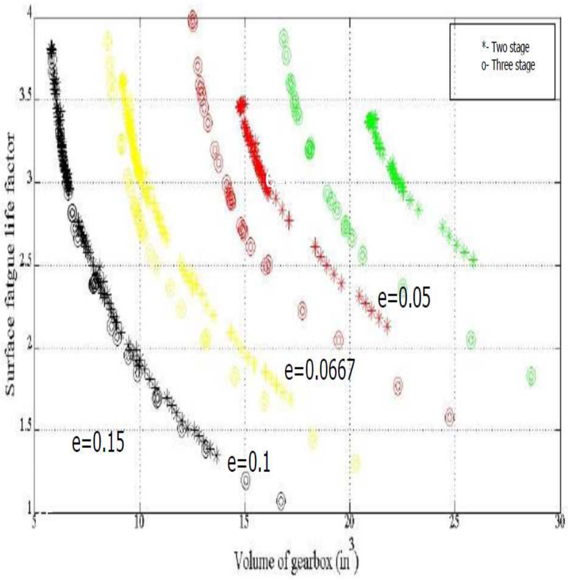
$$\sigma_H = 2.3 \times 0.79$$

$$\sigma_H = 1.81$$

5. Results and Discussions:

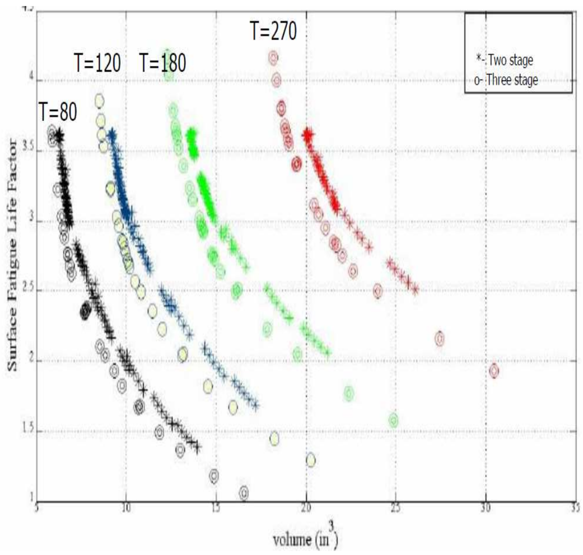
We present the Pareto optimal curves generated for the multi-objective optimization problem defined earlier. The MATLAB® optimization toolbox

Is used to compute the results. Two cases are studied. In the first case, the input torque (T_{in}) is kept constant and the overall speed ratio (*e*) is varied. The plot in Figure 4.2 shows the Pareto optimal curves for the speed reduction ratios of 0.15, 0.1, 0.0667 and 0.05.



Chat-1: Plot of surface fatigue life factor versus volume by varying the overall speed ratio.

In the second case, the overall speed ratio is kept constant and the input torque (*Tin*) is varied. The Pareto optimal curves generated for this analysis.



Chat-2: Graph of Surface fatigue life versus number of cycles.

From the two figures we can draw some conclusions. For the case in which input torque (*Tin*) is kept constant and overall speed ratio is varied, we observe that for lower speed ratio there is no gain in using a three-stage gearbox in place of a two stage one. Also, as the speed ratio increases, the weight of the gearbox increases. For higher speed ratio, the gain in weight reduction is significant for three-stage over two-stage gearbox. For a three stage gearbox, the overall speed ratio gets divided over three stages and so the stresses generated are lower as compared to those generated in a two stage gearbox. This reduces the gear diameters required for each stage and consequently the overall weight of gearbox.

6. Conclusion:

1. The proposed work illustrates briefly about the use of multi criteria optimization method which can be implemented as a component of an expert system for gear train design.
2. The non linear optimization method provides a systematic and step-by-step procedure of formulating the problem and generating the Pareto solution set.
3. The implementation of non linear optimization method is shown through a gear-train design problem.
4. The gear parameters are taken and design to helical gear and find out to speed ratio values.
5. The Pareto optimal solution of MAT lab is used to find out the difference of two stage and three stage gear box of torque and speed ratio.
6. The non linear optimization method is very important design process. In this method improve the efficiency, quality and reduce the cost.

7. References:

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