MULTI OBJECTIVE OPTIMIZATION OF BEVEL GEAR DESIGN USING REAL CODED GENETIC ALGORITHM

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ABSTRACT

Optimization plays an important role in many engineering applications. In design activity, an optimal design is achieved by comparing a few alternative design solutions created by using prior problem knowledge. In such activity the feasibility of each design solution is first investigated. Thereafter an estimate of the underlying objective of each design solution is computed and the best solution is adopted. Optimization algorithms provide systematic and efficient ways of creating and comparing new design solutions in order to achieve an optimal design. In this paper one type of Genetic Algorithm, Real Coded Genetic Algorithm (RCGA) is used to optimize the design of Bevel gear pair and a combined objective function with maximizes the Power, Efficiency and minimizes the overall Weight, Centre distance. The performance of the proposed algorithms is validated through LINGO Software and the comparative results are analyzed.

Key words: Bevel Gear, Gear pair design, Multi-Objective Optimization, LINGO, Real Coded Genetic Algorithm.

NOTATIONS

GA	:	Genetic Algorithm
RCGA	:	Real Coded Genetic Algorithm
P	:	Power transmitted in kW
Hs	:	Specific sliding velocity at start of approach action
Ht	:	Specific sliding velocity at end of recess action.
i	:	Gear (or) transmission ratio
z_1, z_2	:	Number of teeth in pinion, gear
d_1, d_2	:	PCD of large end of pinion, gear in mm
Ro	:	Outside radius of large end of bevel gear in mm
R_2	:	Pitch radius of large end of bevel gear in mm
Ro	:	Outside radius of large end of bevel Pinion in mm
r	:	Pitch radius of large end of bevel Pinion in mm
ρ	:	Density of the material in kg/mm^3
E	:	Young's modulus in N/mm^2
m_t	:	Transverse Module in mm
σc	:	Induced crushing stress in N/mm^2
$[\sigma c]$	:	Allowable crushing stress in N/mm^2
σb	:	Induced bending stress in N/mm^2
$[\sigma b]$	:	Allowable bending stress in N/mm^2
b	:	face with of gear and pinion in mm
R	:	Cone distance in mm.
P_L	:	Percent power loss
$[Mt]$	:	Design twisting moment in N mm
η	:	Efficiency, %
y	:	Form factor
f	:	Average coefficient of friction
Φn	:	Normal pressure angle in degrees
θ	:	Pitch cone angle of bevel gear
γ	:	Pitch cone angle of bevel pinion

I. INTRODUCTION

Bevel gears are used to transmit power between two intersecting shafts and have straight or spiral teeth. Teeth are usually cut at an angle so that the axes intersect at 90°, but any other angle may be used. Bevel gears are not interchangeable and are designed in pairs. Since these gears are cut on conical surfaces, the height of tooth will not be uniform. The height of tooth is maximum at outer and minimum at inner part. The face of the teeth converges at a point called apex. This is a common point for the two mating bevel gears and also this is the point of intersection of the gears. There are large numbers of variables involved in the problem of optimization of bevel gear. Therefore it is very difficult to solve. If more than one functional objectives are considered and it requires a multi objective optimization.

Jean-Luc Marcelin [1] have evaluated with the integrated optimization of mechanisms with genetic algorithms and the possible use of neural networks for complex mechanisms or processes. Heike Trautmann et al [2] have developed statistical methods to help modelling the complex response surfaces for real-world problems such as design and optimization problems in mechanical engineering. Kalyanmoy Deb and Jain [3] has proposed a non-sorted Genetic Algorithm II for optimizing multi speed gear box which consider multi objective such as maximizing the power and minimizing the total volume of the gear.

Marco Barbieri et al [4] have discussed spur gear noise reduction using geometrical modifications are compared. A genetic algorithm is proposed to find the best solutions inside the parameters space. KWON Soon-man et al [5] have discussed, in hypotrochoidal gear pump for rotor wear design, genetic algorithm was used as an optimization technique for minimizing the wear rate proportional factor. Shantanu Gupta et al [6] have evaluated, three primary objectives for a rolling bearing, dynamic capacity, static capacity and hydro dynamic minimum film thickness have been optimized separately and simultaneously with non-dominated sorting based genetic algorithm II. Sylvester V. Ashok et al [7], this paper evaluates the parameters and design options such as gear type, diametral pitch, material, helical angle, shafting and overall configuration with genetic algorithm. Sanchez Caballero V et al [8] have developed genetic algorithm for optimizing transmissible power, reduction ratio etc, for a cylindrical parallel gear trains.

M. Heidari et al [9], this paper discusses the use of variable input speed for optimization of kinematic characteristic of Geneva mechanism. The objective is lowering maximum acceleration and jerk of driven wheel. Zhou et al [10] proposed an ant colony algorithm to solve the prematurity and unsteadiness problem in GA for job shop scheduling with the objective of minimization makespan. Mian Li et al [11] have developed a new Robust Multi-Objective Genetic Algorithm (RMOGA) that optimizes two objectives, a fitness value and a robustness index on multi-objective engineering design optimization problems. Tae Hyong Chong et al [12] have proposed optimization of weight (size) problem of the two-stage gear train and the simple planetary gear train using genetic algorithm. With design parameter such as strength, durability, interference, contact ratio, etc. T. A. Antal [13] has developed a new algorithm for helical gear design with addendum modification. S Padmanabhan et al [14] have evaluated worm and worm wheel gear pair with multi objectives such as maximizing power, efficiency and minimizing weight, center distance using metaheuristic algorithms. L. Tudose et al [15] have developed a two-phase evolutionary algorithm for two-stage speed reducer with the objective function as the volume bounded by the inner surface of the reducer housing.

The design optimization of bevel gear is difficult to solve because it involves multiple objectives and large number of design variables. Hence more reliable and robust optimization technique will be helpful in obtaining optimized results for the bevel gear design problems. This paper has made an attempt to use the potential of RCGA and LINGO to solve the Bevel gear design with maximum the Power, Efficiency and minimum the overall Weight, Centre distance as the objective.

II. DESIGN OPTIMIZATION OF BEVEL GEAR PAIR

In this section, one design problem has been considered for the testing the effectiveness of objectives like minimization of the weight and centre distance and maximization power and efficiency of the bevel gear. The design problem considered as, a bevel gear drives to transmit 4kW with the Speed Ratio 4. The input shaft speed is 225 rpm, with non-reversible.

This section, the mathematical models used in bevel design optimization problem are objective functions, simplified form of objective functions, constraints, simplified form of constraints and complete problem [16, 17] are to be discussed.

A.. Objective functions for Bevel Gear Pair:

The objective functions considered in this bevel gear are given below:

Maximization of power transmitted by bevel gear pair. Eqn. (1) represents this objective function.

$$f_1 = P \text{ where, } P^{(L)} \le P \le P^{(U)} \quad [1]$$

Minimization of weight of the bevel gear pair. Eqn. (2) represents this objective function.

$$f_2 = \text{Weight}$$
$$= 9.24 \times 10^{-6} [0.2838 b^3 - 1.762\, m_1 Z_1 b^2 + 3.6369\, m_1^2 Z_1^2 b] \quad [2]$$

Maximization of efficiency of gear pair. Eqn. (3) represents this objective function.

$$f_3 = 100 - P_L \quad [3]$$

P_L = Power loss and it is expressed by the eqn (4)

$$P_L = 50\, f \times \left(\frac{\cos\theta + \cos\gamma}{\cos\phi_n} \right) \times \frac{(H_s^2 + H_t^2)}{(H_s + H_t)} \quad [4]$$

H_s and H_t are calculated by the eqns. (5) and (6) respectively.

$$H_s = i + 1 \left(\sqrt{\left(\frac{R_0}{R}\right)^2 - \cos^2\phi_n} \right) - \sin\phi_n \quad [5]$$

$$H_t = \frac{i+1}{i} \left(\sqrt{\left(\frac{r_0}{r}\right)^2 - \cos^2\phi_n} \right) - \sin\phi_n \quad [6]$$

$R_0 = R +$ one addendum

One addendum for 20° full depth involute system = one average module $= m_{av}$

Where,

m_{av} = average module of gear and pinion

$$r_o = r + m_{av}$$

$$R_O = R + m_{av}$$

$$t_o = \frac{d_1}{2} + m_{av}$$

$$r = \frac{d_1}{2}$$

d_1 = Pitch diameter of the large end of bevel pinion in mm $= m_t Z_t$

$$R_o = \frac{d_2}{2} + m_{av}$$

$$R_2 = \frac{d_2}{2}$$

d_2 = Pitch diameter of the large end of bevel geat in mm $= m_t Z_2$

Minimization of cone distance of gear pair. Eqn. (7) represents this objective function.

$$f_4 = R = 0.5\, m_t Z_1 \sqrt{i^2 + 1} \quad (7)$$

B. Design Constraints for Bevel Gear:

The constraints considered are bending stress (8), crushing stress (10), gear ratio (12), cone distance between bevel pinion and gear (13), number of teeth in pinion (15) and module (16). The mathematical models for constraints are formulated and given below. Equations (9), (11), (14) and (17) have been adopted from [17]:

(a) Bending stress

$$\sigma_b \le [\sigma_b] \quad [8]$$

$$\sigma_b = \frac{R\sqrt{(i^2+1)}\,[M_t]}{(R-0.5b)^2\,bmy} \times \frac{1}{\cos\Phi_n} \quad [9]$$

(b) Crushing stress

$$\sigma_c \le [\sigma_c] \quad [10]$$

$$\sigma_c = \frac{0.72}{R-0.5}\sqrt{\frac{(i^2 \pm 1)^3}{ib}\,E\,[M_t]} \quad [11]$$

(c) Gear Ratio

$$i = 4 = \frac{Z_2}{Z_1} \text{ (or) } \frac{d_2}{d_1} \quad [12]$$

(d) Cone distance

$$R \ge R_{\min} \quad [13]$$

Where,

$R_{\min} =$ Minimum cone distance calculated by the formula based on surface Compressive stress.

The minimum cone distance is represented by the eqn. (14).

$$R_{\min} = \psi\sqrt{i^2+1}\,\sqrt[3]{\left(\frac{0.72}{(\psi_y - 0.5)\,[\sigma_c]}\right)^2 \frac{E\,[M_t]}{i}} \quad [14]$$

(e) Number of Teeth

The number of teeth must be integer:

$Z_i \,\varepsilon\, I$, for $i =$ 14, 15, 16, 17, 18, 19, 20 [15]

(f) Module

$$m_{av} \ge m_{av\min} \quad [16]$$

Where,

$m_{av} =$ average module

$m_{av\min} =$ minimum average module calculated by the formula based on bending stress

The minimum average module '$m_{av\min}$' is represented by the eqn. (17).

$$m_{av\min} = 1.28\,\sqrt[3]{\frac{[M_i}{y\,[\sigma_b]\,\psi_m Z_1}} \quad [17]$$

$\Psi_m =$ Ratio between the face width and the average module to calculate the value of '$m_{av\min}$.

C. Complete Design Problem for Optimization:

The complete bevel gear design problem is stated below.

(i) Maximize $f_1 = P$ where, $P^{(L)} \le P \le P^{(U)}$ [18]

(ii) Minimize

$$f_2 = 9.24 \times 10^{-6}\,[0.2838\,b^3 - 1.762\,m_1\,Z_1\,b^2 + 3.6369\,m_t^2\,Z_1^2\,b] \quad [19]$$

(iii) Maximize $f_3 = 100 - P_L$ [20]

$$P_L = 5.16 \times \frac{(H_s^2 + H_t^2)}{(H_s + H_t)} \quad [21]$$

$$H_s = 5 \times \left\{\left[\left(1 + \frac{0.4374}{Z_1}\right)^2 - 0.883\right]^{0.5} - 0.342\right\} \quad [22]$$

$$H_t = 1.25 \times \left\{\left[\left(1 + \frac{1.749}{Z_1}\right)^2 - 0.883\right]^{0.5} - 0.342\right\} \quad [23]$$

(iv) Minimize $f_4 = 2m_t Z_1$ [24]

Subject to,

$$P\,Z_1^2\,(m_t Z_1 - 0.5b)^{-2}\,b^{-1}\,(Z_1+5)^{-1} \le 0.42955 \times 10^{-3} \quad [25]$$

$$P^{0.5}\,b^{-0.5}\,(2\,m_t Z_1 - 0.5\,b)^{-1} \le 0.003351 \quad [26]$$

$$m_t Z_1\,P^{-0.333} \ge 38.71 \quad [27]$$

$$m_t^3\,(Z_1+5)\,P^{-1} \ge 88.3795 \quad [28]$$

$$Z_2 = i\,Z_1 \text{ (or) } Z_2 = 4\,Z_1 \quad [29]$$

Number of teeth in pinion is 13, 14, 15, 16, 17, 18, 19 and 20

D. The Input Parameters:

The parameters considered for the bevel gear pair design is given below.

Material of gear and pinion $= 40Ni2Cr1Mo28$

Density of the material (ρ) $= 8.836 \times 10^{-6}$ kg/mm^3

Gear ratio (i) $= 4$

Allowable bending stress $[\sigma_b]_{at} = 400$ N/mm^2

Allowable crushing stress $[\sigma_c]_{at} = 1100$ N/mm^2

Input speed $= 225$ rpm

Young's modulus of the material (E)

$= 2.15 \times 10^5$ N/mm^2

Normal pressure Angle (Ψ_n) $= 20°$

Co-efficient of friction (f) $= 0.08$

Product of Load Concentration Factor (k) and Dynamic Load Factor (k_b) $= 1.3$

Ratio between cone distance and face width (ψ_y) to calculate the value of '$R_{\min}$' $= 4$

Ratio between face width and average module (ψ_m) to calculate the value of

'$m_{av} \min = 10$'

a. Variable Bounds:

- Transverse module is varied from 6 to 7 mm and 7 to 8 mm in the range of 0.001 mm.
- Face width is varied from 75 mm to 125 mm in the range of 0.001 mm.
- Number of teeth in pinion – 13, 14, 15, 16, 17, 18, 19 and 20.
- Power is varied from 4 to 6kW in the range of 0.001 kW.

E. Proposed Design Objective Function

In this, bevel gear pair design problem has four different parameters are considered i.e., power, weight of material, efficiency and center distance. Since all these parameters are on different scales, these factors are to be normalized to the same scale. For maximizing criterion value, the values are normalized by dividing its value with the normalizing factor, $\max_i$, which is the maximum value of this criterion obtained from the solutions that have been explored so far and for a minimizing criterion value, it is normalized by dividing the normalizing factor, $\min_i$, with its value.

The normalized objective function is obtained as follows:

$$COF = \sum_{i=1}^{n} NW_i \times N(X_i) \quad [30]$$

Where,

$COF =$ Combined objective function

$W_i =$ pre normalized weight of criterion i.

$NW_i =$ normalized weight of criterion i.

$$\text{Where } NWi = \frac{W_i}{\left(\sum_{i=1}^{n} W_i\right)} \quad [31]$$

$N(X_i) =$ normalized value of criterion i of solution X.

Where,

$$N(X_i) = \frac{X_i}{\max_i} \text{ for maximizing criterion.} \quad [32]$$

$$N(X_i) = \frac{\min_i}{X_i} \text{ for minimizing criterion.} \quad [33]$$

$X_i =$ pre normalized value of criterion X.

$\max_i =$ pre normalized maximum value of criterion i among all solutions explored so far.

$\min_i =$ pre normalized minimum value of criterion i among all solutions explored so far.

$N =$ number of criteria.

Hence the COF for this problem is,

$$COF = \left[\left(\frac{\text{power}}{\text{max. power}} \times NW_1\right) + \left(\frac{\text{min} \cdot \text{weight}}{\textit{weight}} \times NW_2\right) + \left(\frac{\text{efficiency}}{\text{max. efficiency}} \times NW_3\right) + \left(\frac{\text{min} \cdot \text{cent} \cdot \text{dist}}{\text{cent} \cdot \text{dist}} \times NW_4\right)\right] \quad [34]$$

Where NW_1, NW_2, NW_3 and $NW_4 = 0.25$

III. METHODOLOGY FOR MULTI OBJECTIVE OPTIMIZATION

After the extensive literature survey, many researchers have shown more interests on non-traditional optimization techniques such as Genetic Algorithm, Simulated Annealing, etc and applied the same in various engineering fields [18-21]. Here, an attempt to be made with a new Real Coded Genetic Algorithm (RCGA) and LINGO software tool for the design of bevel gears.

A. Real Coded Genetic Algorithm:

In this work one type of Genetic Algorithm, Real Coded Genetic Algorithm (RCGA) is used. The RCGA uses Mixed Integer Representation (MIR) for representing the control variables, Tournament selection, Simulated Binary Crossover and Polynomial Mutation.

a. Representation:

Mixed integer representation is used for the control variables. The module, thickness, number of teeth in pinion, the power, maximum power, minimum weight, maximum efficiency, minimum cone distance and COF are represented in a control string. Module, thickness of gear pair & power are represented as continuous variables within limits. The number of teeth is represented as discrete variable. The control string will be as per the eqn. (35).

$$X = [m_t, b, z, p, f_1, f_2, f_3, f_4, COF] \qquad [35]$$

b. Tournament Selection:

The tournament selection provides a selective pressure by holding a tournament competition among individuals. The best individual (the winner) from this group is selected as parent. That is, any two strings are randomly selected from this population and the COF value is compared. The string having the lowest COF will be stored in the new mating pool.

For example if the two strings are selected randomly, lower COF string is stored in the new mating pool. If any string having the lowest COF is selected more than one time, it will be stored that much time in the pool. This process is repeated until the mating pool for generating new offspring is filled. Tournament selection is used as selection mechanism in order to avoid premature convergence.

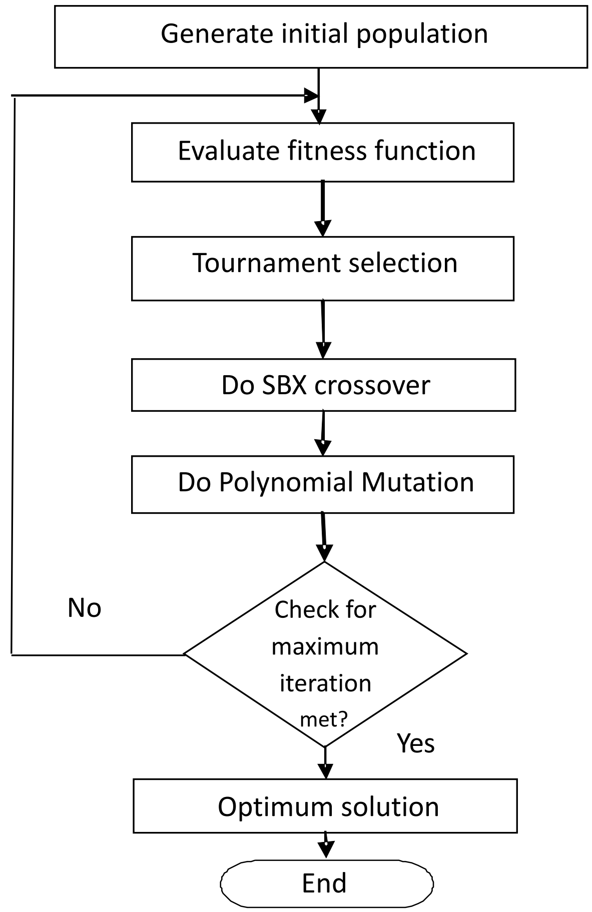

Fig. 1. The Algorithm for GA

c. Simulated Binary Crossover

The simulated binary crossover performs the crossover variable-wise using SBX operator. It creates children solutions in proportion to the difference in parent solutions.

The following steps to create two children solutions from two parent are given below.

1. Choose a random number $u_i = \in [0, 1]$
2. Calculate β_{qi} as given in eqn. (36)

$$\beta_{qi} = \begin{cases} (2u_i)^{\frac{1}{n_c+1}}, & u_1 \leq 0.5 \\ \left(\frac{1}{2\,(1-u_i)}\right)^{\frac{1}{n_c+1}}, & \text{otherwise} \end{cases} \qquad [36]$$

where β_{qi} is the spread factor and is defined as the ratio of the absolute difference in offspring values to that of the parents. η_c is the crossover index of SBX operator.

3. Then compute the offspring $x_i^{(1,\, t+1)}$ & $x_i^{2,\, t+1}$ as given in the eqn. (37),

$$x_i^{(1,\, t+1)} = 0.5\,[(1+\beta_{qi})\, x_i^{(1,\, t)} + (1-\beta_{qi})\, x^{(2,\, t)}$$

$$x_i^{(2,\, t+1)} = 0.5\,[(1+\beta_{qi})\, x_i^{(1,\, t)} + (1-\beta_{qi})\, x^{(2,\, t)} \qquad [37]$$

To carryout the SBX operation any one of the variables m_t, b, Z and p may be selected from the above said parents.

d. Polynomial Mutation:

Newly generated offspring undergo polynomial mutation. Like in the SBX operator, the probability distribution can also be a polynomial function, instead of a normal distribution. The new offspring $y_i^{(1,\, t+1)}$ is determined as follows,

$$Y_i^{(1,\, t+1)} = X_i^{(1,\, t+1)} + (x_i^U - x_i^L)\, \overline{\delta_i} \qquad [38]$$

x_i^U and x_i^L are the upper and lower limit values.

Where, the parameter $\overline{\delta_i}$ is calculated from the polynomial probability distribution.

$$P(\delta) = 0.5\,(\eta_m + 1)\,(1 - |\delta|)^{\eta_m}$$

$$\overline{\delta_i} = \begin{cases} (2r_i)^{1/(\eta_m+1)} - 1, & \text{if } r_i < 0.5 \\ 1 - [2\,(1-r_i)]^{1/(\eta_m+1)}, & \text{if } r_i \ge 0.5 \end{cases} \qquad [39]$$

Where, η_m is the mutation index. In this operator the shape of the probability distribution is directly controlled by the external parameter η_m and distribution is not dynamically changed with generations.

e. Stopping criteria:

The approach followed in this work, is to stop the computation after reaching the required number of iterations. The maximum number of iterations adopted here is 10. For each iteration the population is generated continuously by 200 times.

B. NON LINEAR PROGRAMMING (NLP) IN LINGO

LINGO is a simple tool for utilizing the power of linear and non-linear optimization to formulate large problems concisely, solve them, and analyze the solution. In this paper the models have been solved with the help of LINGO provided by LINDO SYSTEMS Inc. Chicago. An optimization LINGO model consists of three parts. They are objective function, variables and constraints. Once the LINGO model has been created and entered into the LINGO model window, the model can be solved.

In this work four models have been created in LINGO to solve the four objective functions. As these models have been solved separately, these values can not be taken as optimum values. Hence a fifth model has been created in LINGO which combines all the four objective functions. The minimum values of weight and centre distance and maximum values of power and efficiency obtained from the previous models have been given as inputs for the fifth model and *COF* value and corresponding optimal results have been found out.

IV. RESULTS AND DISCUSSION

The Mathematical models for the design problem have been formulated in terms of design variable *m*, *b*, *z* and *P* discussed in the section 2.3. Initially, the input values are generated randomly with their variable bounds (section 2.4.1). If the generated values satisfy the design constraints (section 2.2), then the values of objective functions *f*1, *f*2, *f*3 and *f*4 are computed along with *COF*. The optimum values of objective function and design variables corresponding to the minimum *COF* value obtained by the proposed algorithms for the test problem are shown in a Table 1.

Table 1. Optimum results for the for Bevel Gear Pair

Tool	Transverse Module (mm)	Thickness (mm)	No. of Teeth in Pinion	Power (kW)	Weight (kg)	Efficiency (%)	Cone Distance (mm)
LINGO	7	125	20	4	51.8394	95.9988	280
RCGA		123.822	20	4.251	51.588	95.9988	280
LINGO	8	125	20	4	71.9554	95.9988	320
RCGA		122.734	20	4.312	71.9470	95.9988	320

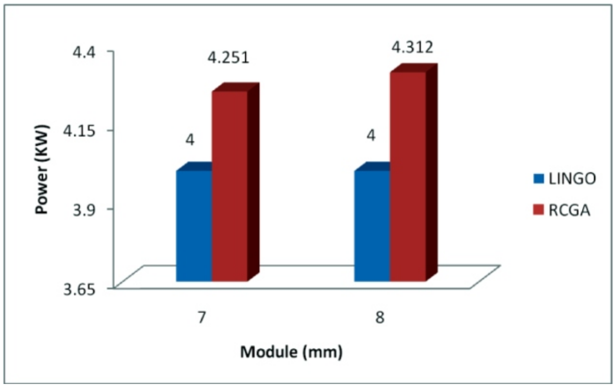

Fig. 2. Comparion of Power

The bevel gear test problem is carried for the module range 6 mm to 8 mm and the bevel gear design optimization carried by the RCGA and LINGO software. From the optimum results, power is compared for RCGA and LIGO and the chart is shown in the figure 2. It is clearly understood from the figure 2, RCGA gives the maximum Power comparing with LINGO. The Power gained by 6.25% and 7.8% with respect to 7 mm and 8 mm modules respectively. Similarly, Weight reduced nominally with respect to 7 mm and 8 mm modules. Both the RCGA and LINGO provides the same maximum Efficiency and minimum Cone distance.

V. CONCLUSION

The word optimization is from 'Optimum' which implies a point at which the conditions are best and most favorable. An optimum value may represent a maximum value or a minimum value. In this paper an attempt has been made to obtain optimal solution of bevel gear pair design problem. The various design variables available for a gear pair design, the power, weight, efficiency and center distance have been considered as objective functions and bending stress, crushing stress as vital constraints to get an efficient compact and high power transmitting gear pairs. The proposed new Real Coded Genetic Algorithm (RCGA) has produced persuasive results for the design problems when compared with the LINGO tool. In particular RCGA has shown significant increase in power. As a future work, minimization of gear noise, maximization of life cycle and minimization of transmission error can also be included in the objective function to obtain a more reliable gear pair design.

REFERENCES

[1] Jean-Luc Marcelin, 2010, "Integrated Optimization of Mechanisms with Genetic Algorithms", Engineering, 2, 438-444.

[2] Heike Trautmann, Jorn Mehnen, 2009, "Statistical Methods for Improving Multi-objective Evolutionary Optimisation", International Journal of Computational Intelligence Research. Vol.5, No.2, pp. 72–78.

[3] Deb, K.; Jain, S., 2003, "Multi-Speed Gearbox Design Using Multi-Objective Evolutionary Algorithms", Journal of Mechanical design, Vol.125. 609-619.

[4] Marco Barbieri, Giorgio Scagliarini, Giorgio Bonori, Francesco Pellicano and Gabriele Bertacchi, 2008, "Optimization Methods For Spur Gear Dynamics", ENOC 2008, Saint Petersburg, Russia.

[5] KWON Soon-man, KIM Chang-Hyun, SHIN Joong-ho, 2011, "Optimal rotor wear design in hypotrochoidal gear pump using genetic algorithm", J. Cent. South Univ. Technol. 18: 718-725.

[6] Shantanu Gupta, Rajiv Tiwari, Shivashankar B. Nair, 2007, "Multi-objective design optimisation of rolling bearings using genetic algorithms", Mechanism and Machine Theory 42, 1418–1443.

[7] Sylvester V. Ashok, Daniel P. Schrage and Alexander Robledo, 2011, "A Systems Engineering Modeling and Simulation Approach for Rotorcraft Drive System Optimization", American Helicopter Society 67th Annual Forum, Virginia Beach, VA.

[8] Sánchez Caballero, V. Colomer Romero, R. Pla Ferrando, A.V.and Martínez Sanz, S., 2008, "Cylindrical parallel gear trains design by means of Genetic optimization", Selected Proceedings from the 12th International Congress on Project Engineering". Zaragoza.

[9] Heidari M., Zahiri M., and Zohoor H., 2008, "Optimization of Kinematic Characteristic of Geneva Mechanism by Genetic Algorithm", World Academy of Science, Engineering and Technology, pp 387-395.

[10] Zhou P, Li X-P, Zhang H-F, 2004, "An ant colony algorithm for job shop schedulingproblem", Proceedings of the 5th worldcongress on intelligent control and automation, China,15–19, pp 2899–2903.

[11] Mian Li, Shapour Azarm and Vikrant Aute, 2005, "A Multi-Objective Genetic Algorithm for Robust Design Optimization", GECCO'05, Washington, DC, USA. pp. 771-778

[12] Tae Hyong Chong, Joung Sang Le, 2000, "A Design Method of Gear Trains Using a Genetic Algorithm", International Journal of the Korean Society of Precision Engineering, Vol. 1, No. 1,pp 62-70.

[13] Antal T.A., 2009, "A new algorithm for helical gear design with addendum modification", MECHANIKA. No.3(77), pp 53-57.

[14] Padmanabhan. S, Chandrasekaran M. and Srinivasa Raman V., 2010, "Optimization Of Worm And Worm Wheel Design Using Metaheuristic Algorithms", IETECH Journal of Mechanical Design, Vol: 4, No: 2, pp 043 – 047.

[15] Tudose L., Buiga O., Jucan D. and ªtefanache C., 2008, "Optimal design of two-stage speed reducer using two-phase evolutionary algorithm", International Journal Of Mechanics, Issue 3, Volume 2, 2008, pp 55 – 66.

[16] Dudley, D.W., 1954, Practical Gear Design, McGraw –Hill Book Co, New York.

[17] Design Data Book, Faculty of Mechanical Engineering, PSG College of Technology, Coimbatore-641004, Tamilnadu, India.

[18] Baskar N, Asokan P, Saravanan R, Prabhakaran G, 2005, "Optimization of machining parameters for milling operations using non-conventional methods", International Journal of Advanced Manufacturing Technology, Vol. 25, pp 1078-1088.

[19] Asokan P, Saravanan R., Vijayakumar K, 2003, "Machining parameters Optimization for Turning Cylindrical stock into a continuous Finished Profile using Genetic Algorithm and Stimulated Annealing", International Journal of Advanced Manufacturing Technology, Vol. 21, pp 1-9.

[20] Saravanan R., Sachithanandam, M., 2001, "Genetic Algorithm for multi variable surface grinding process optimization using a multi objective function mode". International Journal of Advanced Manufacturing Technology, Vol 17, pp 330 - 338.

[21] Vijayakumar K, Prabhaharan G, Asokan P, Saravanan R, 2003, "Optimization of multi-pass turning operations using ant colony system", International Journal Of Machine Tool & Manufacture, Vol.43, pp 1633-1639.