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Advanced Multi Criteria Optimal Design of Spiral Bevel Gear Pair using NSGA – II

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Abstract

In gear applications, quality of design significantly influences transmission, machine performance, size and weight of the gears. In the present work, a nonlinear optimization problem having three objective functions, five design variables and eleven constraints considering a spiral bevel gear pair is solved. The aim of this research is to optimize weight, pitch cone distance, and efficiency by formulating three cases. In Case 1, the objective functions, namely, weight and pitch cone distance are minimized, while treating efficiency as constraint. In Case 2, the objective functions weight is minimized and efficiency is maximized, keeping pitch cone distance as constraint. In Case 3, the objective functions pitch cone distance is minimized and efficiency is maximized, having weight as constraint. Pareto frontiers are generated by Non-dominated Sorting Genetic Algorithm (NSGA-II). Simulation is analysed and validated with literature. Results show that there is a considerable rise in weight, module, and efficiency and a decrease in cone distance than literature. Results also indicate that Case 2 formulation offers the best optimal design parameters.

Keywords: Spiral Bevel Gear Drive; Design Optimization; Critical constraints; NSGA-II.

1. Introduction

Bevel gear drive is applied wherever change of directions is desirable in transmission. Spiral bevel gears are one of the basic mechanical units to transmit motion between concurrent axes. As they offer great concurrence and even transmission, they are extensively used in the aerospace, automotive and large mechanical transmission systems[1].They yield smoother operation, less noise and vibration, since they have big overlapping tooth action. They also can carry more loads, as they possess evenly distributed tooth loads. A spiral bevel gear pair is shown in Figure 1.

Design optimization of gear transmission systems has been a puzzling problem to researchers for several years because of the following reasons: a) The practical gear design is characterized by many design parameters, much calculation time, and error susceptibility. b) It requires repetitive calculation, interrogation and drawings for gear design which leads to additional effort. Nevertheless, the use of latest computers through intelligent techniques, aid us to solve gear optimisation problems handily[2].

Optimization problems in gear design involve multiple objective functions. As multi criteria optimization offers pareto-optimal solutions set to the choice of a decision maker, it is suitable for gear research [3]. In such optimization, weights are also allowed to make a trade-off between criteria. It is highly important to identify a set of Pareto optimal solutions which satisfy all the objectives as better as possible.

Figure 1 A spiral bevel gear pair

The current advances in the research on design optimization of bevel gears is as follows:

Emmanuel Mermoz, et. al [4]optimized a spiral bevel gear using Finite Element Method (FEM),replacing sensitivity analysis. They used optimization algorithms to automatically compute the tooth contact flanks surfaces. Tetsu Nagata, et.al [5] designed tooth contact analysis and tooth flank form measurement technique to calculate meshing condition by considering large spiral bevel gears. Faydor et. al[6]improved bearing contact to achieve a predesigned parabolic function, so as to reduce magnitude of transmission errors. Liang and Xin [7] specified spiral gear mesh through dynamic simulation approach. They

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calculated angular speed, torque and meshing force on the curves of spiral bevel gears. Chung –Yunn et.al [8] performed dynamic simulation of the spiral bevel gears with specified mesh. They got the curves of angular speed, torque and meshing force on the spiral bevel gears through simulation in order to reveal dynamic characteristics of gear driving device. Chandrasekaran et al. [9] presented a brief review and analysis of latest advancements in bevel gears optimization research. Chandrasekaran et al[10]optimized a design of spiral bevel gear pair considering efficiency, weight and cone distance subject to mechanical constraints using NSGA-II.

Bevel gear optimisation through non-linear formulations, metaheuristics and other algorithms are as follows:

Arunachalam,et al [11]maximised power, efficiency and minimized overall weight and centre distance taking into account a combined objective function. They used LINGO and Real Coded Genetic Algorithm (RCGA), Ant Colony Optimisation (ACO) and Particle Swarm Optimisation (PSO) algorithms for solving the problem. Zhang, et al [12]used optimization design software Visual Basic (VB) for straight bevel gear design based on MATLAB, and Genetic Algorithm (GA). They used augmented penalty function and integer serial number encoding to obtain global optimal solution. Rai et al [13], minimized the volume of straight bevel gear considering scoring failure in straight bevel gear by advanced particle swarm optimization and RCGA. Li Tiejung [14]solved a variable helix angle bevel gear problem through a mathematical model based on GAs. They compared the same with traditional optimization methods of literature, and showed that Gas are reasonable to solve gear application problems with many constraints and variants. Padmanabhan. et.al. [15] formulated a combined objective function to optimize bevel gear pair design based on RCGA and LINGO. Ebenezer et al [16] proposed nature inspired algorithms, namely, Simulated Annealing (SA), Fire fly Algorithm (FA) and Cuckoo Search (CS) to optimize volume of a straight bevel gear. Zolfagari et al. [17]optimized volume of straight bevel gear based on the American Gear Manufacturers Association (AGMA)stipulations using SA and GA.

Bevel gear optimisation taking into account power losses are as follows:

Graham Johnson et al [18].optimized design of shrouded meshing gears pair that run at representative speeds and oil flow in a rig with speed and torque measurement. They quantified three main windage power loss contributors, namely, air alone, recirculation of oil under the shroud and acceleration of the feed oil. Oil and bearing churning losses, seal and windage losses are hooked on the speed of the gear pair. Bearing sliding friction and rolling friction losses are reliant on the load on the gear pair.To measure air losses (windage losses) researchers in the past conducted experiments on test rigs. Diab et. al [19] conducted a set of experiments in a test rig and obtained an empirical equation for determining windage power loss. They considered dimensional and fluid flow analysis.

The cited literature reveals a clear gap as follows:

1. Certain essential constraints, namely, contact ratio, line of action, load carrying capacity and power loss due to meshing loss in bearings and seals were not considered. Contact ratio is the main factor in the load carrying capacity and dynamic performance of the gears. Greater contact ratio lowers the vibration and noise of the transmission system. But in this research, all these are considered.
2. The researchers dealt with only two design variables, namely, number of teeth of pinion and module of the gear. In this paper, three more additional design variables viz. shaft diameter, power and face width are included.
3. Some researchers considered only combined objective functions. In such cases, the decision maker does not know how to choose the weighing factors, when functions of optimization problem are not familiar to him[20].In this research, multiple objectives are formulated as three cases, and in each case, two objectives are taken for optimization with the third one as a constraint along with additional constraints, which does not demand weighting factors.
4. Only heuristic algorithms or conventional techniques were generally used by the researchers in the literature. As these algorithms partition the whole problem into sub problems, they are appropriate only for solving single criteria gear design problems. As multi-criteria problems are computationally intensive, it is fair to use multi- objective optimization algorithms, such as NSGA-II, which gives greater performance and does not require any weight functions.

Taking cognizance of the above observation, to bridge the gap besides normal constraints, certain other additional critical constraints and design variables are also considered. This work is based on Chandrasekhar et al. [10]and Arunachalam,et al [11], but in this work, three more design variables, six other additional constraints are incorporated for optimization in three separate cases to augment performance, design and power transmission.

2. Design optimization formulation of the Spiral bevel gear problem

In this research, a spiral bevel gear pair is considered. The design data for the problem are given in Table 1.

2.1. Design example

A spiral bevel gear pair presented in [10] is considered. Certain additional design variables and critical constraints are also taken into account.

2.1.1. Objective Functions

The aim is to optimize objective functions, namely, weight, pitch cone distance and efficiency of the spiral bevel gear pair. To decide optimum solutions, three cases based on the problem's objectives are formulated separately and optimized along with additional constraints.

In Case 1, the objective functions weight and pitch cone distance are minimized, with efficiency as constraint, in Case 2, the objective functions weight is minimized and efficiency is maximized, keeping pitch cone distance as constraint and in Case 3, the objective functions pitch cone distance is minimized and efficiency is maximized, having weight as constraint. The limiting values of the objective functions, which are treated as constraint in each case are selected from literature [10].

Table 1 Design data for the bevel gear drive optimization

Parameters	Value	Parameters	Value
Transferred power, (P)	37.285kW	Working temperature (deg C)	50
Material for pinion and gear	Steel 20 Mn 5 Cr 5 IS: :4432-1988	Pinion / gear tooth shape	Properly crowned
gear ratio, i_{tot}	4.778	Gear teeth form	Full depth, conifex
Input pinion speed,n(rpm)	500	Reliability	0.99
Coefficient of friction, f	0.08	Load on driven machine	Medium shock
Pressure angle,ϕ(deg)	20	Shafts material	SAE 1060
Young's Modulus, E	$2.15\times 10^5\ N/mm^2$	Pinion/gear tooth hardness (HB)	350
Ratio between cone distance and face width ,ψ_y	$0.357Z_1$	Life (no of cycles)	10^8
Ratio between average module and face width ,ψ_m	8	Safety factor –shaft design, S_{FS}	1.5
Density of the material	$7.86\times 10^{-6}\ kg/mm^2$	Safety factor for bending, S_F	1.2
Design bending stress, $[\sigma_b]$	$430\ N/mm^2$	Safety factor for pitting, S_H	1.2
Design crushing stress,$[\boldsymbol{\sigma_c}]$	$\mathbf{1100\ N/mm^2}$		

The objective functions equations (1), (2) and (3), are adopted from [10] as follows:

Total weight of the spiral bevel gear pair

$$f_1 = W = W_1 + W_2, \tag{1}$$

where weight of the pinion,

$$W_1 = 42.438\ \rho\ m_t^{\ 3} z_1 \tag{2}$$

and weight of gear,

$$W_2 = 68.52\ \rho\ m_t^{\ 3} z_2 \tag{3}$$

Efficiency of the gear,

$$f_2 = \eta = 100 - P_L \tag{4}$$

where

$$P_L \text{ is Power loss} = 50f\left\{\frac{cos\theta+cos\gamma}{cos\phi_n}\right\}cos^2\beta\,\frac{(H_s^{\ 2}+H_t^{\ 2})}{(H_s+H_t)} \tag{5}$$

and

$$H_s = (i+1)\left\{\left[\sqrt{\left(\frac{R_0}{R}\right)^2 - cos^2\phi_n}\right] - sin\phi_n\right\} \tag{6}$$

$$H_t = \left(\frac{i+1}{1}\right)\left\{\left[\sqrt{\left(\frac{r_0}{r}\right)^2 - cos^2\phi_n}\right] - sin\phi_n\right\} \tag{7}$$

where $R_0 = R + one\ addendum$. One addendum for 20^0 full depth involute system= One average Module $= m_{av}$, where $m_{av} = m_t\left(\frac{\psi_y - 0.5}{\psi_y}\right)$,$r_0 = r + m_{av}$; $R_0 = R + m_{av}$; $r = d_1/2$; $R = d_1/2$, d_1=$m_t z_1$and d_2=$m_t z_2$,d_1, d_2 are pitch diameter of the large end of the bevel pinion and gear in mm, W_1, $W_2$are weight of the pinion and gear.

3. Pitch cone distance of the gear pair,

$$f_3 = R_c = 0.5 m_t z_1\sqrt{i^2+1} \tag{8}$$

2.2. Design variables

The design variable function of the bevel gear pair is formulated as follows:

$$F(x) = F(m, z_1, b, d_s, P) = F(x_1, x_2, x_3, x_4, x_5) \tag{9}$$

where $m$ismodule,z_1 is number of teeth on pinion, b is face width and$d_s$is diameter of the shaft and $P$is input power.

Upper and lower design bound are continuous variables as follows:

$5 \le m \le 10, 5 \le z_1 \le 12; 20 \le b \le 60,\ 15 \le d_s \le 40$ and

$P = 29.828\ kW, 37.285\ kW$ and $52.199\ kW$. Here the design variable power is considered as a discrete variable.

2.3. Constraints

Crucial mechanical constraints along with certain added critical constraints are considered.

2.3.1. Bending stress of the gears

As per design requirements, bending strength and contact stress must be lower than allowable bending stress and contact stress. The equations (10 - 13) are reported in [10].

$$\left[\frac{0.7R\sqrt{(i^2+1)}[M_t]}{(R-0.5b)^2 b m_n Y_v}\right] \le [\sigma_b] \tag{10}$$

where,

R is cone distance (mm) and $[M_t]$ is design twisting torque (Nmm),

$[\sigma_b]$ is allowable bending stress number (N/mm²) , b is face width of gear (mm), $m_n$is normal module (mm),and Y_v is form factor of gear.

2.3.2. Crushing stress of the gears

The contact stress constraint is devised as:

$$\frac{0.72}{(R-0.5b)}\sqrt{\frac{(i^2+1)^3}{ib}E[M_t]} \le [\sigma_c] \tag{11}$$

where i is gear ratio, $E$is Young's Module (N/mm²),$[\sigma_c]$is allowable contact stress number (N/mm²)·$[M_t]$ is design twisting torque (Nmm), and b is face width (mm).

2.3.3. Pitch cone distance

This constraint is developed as:

$$\frac{41.4885}{(0.357Z_1 - 0.5)^{\frac{2}{3}}} \le R \tag{12}$$

where $Z_1$is number of gear teeth and R is cone distance (mm).

2.3.4. Average module

This constraint is established as:

$$1.15cos\beta_{av}\sqrt[3]{\frac{[M_t]}{y_v[\sigma_b]\psi_m z_1}} \leq m_{av} \quad (13)$$

m_{av} is average module(mm), β_{av} , ψ_y are mean spiral angles, y_v is form factor.

2.3.5. Shaft diameter

The equations (14 - 20) are stated in [16].

The shaft diameter constraint is as follows:

$$\left[\frac{32\,S_{FS}}{\pi}\sqrt{\left(\frac{T}{S_y}\right)^2+\left(\frac{M}{S_e}\right)^2}\right]^{\frac{1}{3}} - d_s \leq 0 \quad (14)$$

where $S_y, S_e$are yield strength and endurance limit of the shaft material (N/mm^2),T is torque transmitted by the shaft, (Nmm) and M is maximum bending moment on the shaft (Nmm), $d_s$is shaft diameter (mm) andS_{FS} is safety factor for pitting.

2.3.6. Gear face width

The constraint is laid in equation (15) as follows:

$$b \leq \{0.3R, 10m_t\} \quad (15)$$

where b is gear face width(mm), and m_t is transverse module of gears(mm).

2.3.7. Power loss in the gear

It is an important constraint, as efficiency depends on the power loss of the gear pair.

The range of power loss percentage should be between 1.2 and 2.2% of input power.

It is given by,

$$P_{loss} - 1.2\,\%(P) \leq 0\,, \quad (16)$$

Where

$$P_{loss} = P\,\mu_{mz}H_v + \mu\,Fv + 7.69x10^{-6}d_s^2\,n\,, \quad (17)$$

where P_{loss} is the loss of power (W), F is bearing load (N), v is peripheral speed (m/s), μ_{mz} is average coefficient of friction, H_v is gear power loss factor, μ is coefficient of friction in the bearing, n is rotational speed (rpm),d_s is shaft diameter (mm).

2.3.8. Contact Ratio(CR)

As spiral bevel gears have curved oblique teeth, they mesh with a rolling contact similar to helical gears. So, the action of a spiral gear is same as the helical gear. For such a model bevel gear pair, contact ratio should be between 1.4 and 2. It is given by,

$$1.4 \leq CR \leq 2 \quad (18)$$

where

$$CR = \frac{\left(\sqrt{(r_1+\text{a})^2-r_{b1}{}^2}+\sqrt{(r_2+\text{a})^2-r_{b2}{}^2}-(r_1+r_2)\,sin\phi\right)}{\pi m cos\phi} \quad (19)$$

where suffix 1 for pinion and 2 for gear,$r_b$isbase circle radius (mm),ais addendum (mm), m is module of the gear (mm) and ϕ is transverse pressure angle.

2.3.9. No involute interference

If the pinion tooth makes contact with the gear tooth or the involute of the pinion comes in that range, this occurs. To obtain no involute interference the following constraint is used.

$$\sqrt{(r_1+\text{a})^2-r_{b1}{}^2} - Csin\phi \leq 0 \quad (20)$$

where C is centre distance between the gears and ϕ is transverse pressure angle of the gear.

2.3.10. Load carrying capacity

Load carrying capacity F_1should be more than minimum load carrying capacity F_{min} of the gearas reported in[23]. It is given by,

$$F_{min} - F_1 \leq 0 \quad (21)$$

where

$$F_1 = F_t + \frac{[21(Cb+F_t)]}{21v+\sqrt{(Cb+F_t)}}[25] \quad (22)$$

where F_t is transmitted load(N), $C$is deformation factor depending on machining error, ($C = 228\,e\,Nmm$),$e$is expected error, ($e = 0.02$ mm), v is the velocity of the gearinm /sec and $b =$ face width or base width of gear (mm).

2.3.11. Line of action

To achieve even and constant rotation, arc of action should be more than line of action [1].Accordingly, this constraint is expressed.

$$\frac{2\pi}{tan\phi} \leq z_1 + z_2 \quad (23)$$

where z_1 , z_2 are number of teeth of pinion and gear.

3. Optimization Algorithms

3.1. Multi objective optimization algorithms

Engineering optimization problems can be of single-objective optimization (SOO) or multi-objective optimization (MOO). The SOO is a formulation of a combined function that characterises the overall effect. But the MOO is a construction of multi criteria, which are diverse and conflicting with wide-ranging solution methods. Multi criteria optimization problems along with modelling have been handled by Moneim [26], Benatiallah et al [27], Kazem et al [28] in motion planning and wind systems.

3.1.1. NSGA–II

The NSGA-II algorithm is a modest and direct method as elaborated in [21].It uses an elitist principle, so that only elites can be carried forward to the next generation. It also employs a clear diversity preserving mechanism known as crowding distance. In NSGA- II, the emphasis is on producing non-dominated solutions. It is possible to realise this by using crowded comparison criterion in the tournament selection as well as in the phase of population reduction. In this research, optimization is performed in MATLAB environment using code [22].

4. Results and discussion

Simulation byNSGA -IIfor (Case I, Case 2 and Case 3) respectively are presented in Table 2. Optimal design parameters by the algorithm of this work and the literature [10] are compared.

From Table 2, certain interesting observations are made. The weight of gear has significantly increased from 19.923 kg (forz_1=10) to 22.81 kg (Case 1).It is the same with module also, which has increased from 8.729 mm (for 〖 z〗 _1=10) to 9.853mm(Case 2). But positively, there is a decrease in the value of Cone pitch diameter (186.904 mm) compared to 213.625 mm (for 〖 z〗 _1=10) of the literature. In the same way, there is an increase in efficiency from 97.546% (for 〖,z〗 _1=10) to 98.406% (Case 2).The pinion teeth value has also slightly decreased from 10 (for 〖,z〗 _1=10) to 9.192.A careful observation shows that Case 2 yields better optimal parameters than other cases. The best optimum design parameters (Case 2) are as follows: =9.853mm, 〖 z〗 _1=9.192,b=27.064mm, 〖 d〗 _s=23.84 mm, and W=26.319 kg, R_c =213.625 mm, and η=98.535 % .

Table 2 Optimal values (Case 1, Case 2 and Case 3) by NSGA-II of the present research and literature [10]

Power 37.285kW						
Parameters	Presented paper			Literature		Percentage change in values
	Case 1	Case 2	Case 3	[10]		(between Case 2 and Literature for, z_1=10)
Module (mm), m	9.846	9.853	9.356	8.719	8.241	-13.00
Pinion teeth , z_1	7.788	9.192	8.566	10	11	+0.88
Face width (mm) b	46.335	27.064	24.301	Not considered	Not considered	-
Diameter of the shaft (mm) , d_s	35.952	26.783	30.753	Not considered	Not considered	-
Weight of gear (kg), W	22.181	26.319	19.923	19.923	18.411	-32.10
Cone pitch distance (mm), R_c	186.904	213.625	195.553	213.625	220.978	0.00
Efficiency of the gear (%), η	97.546	98.535	98.406	97.546	97.745	1.01

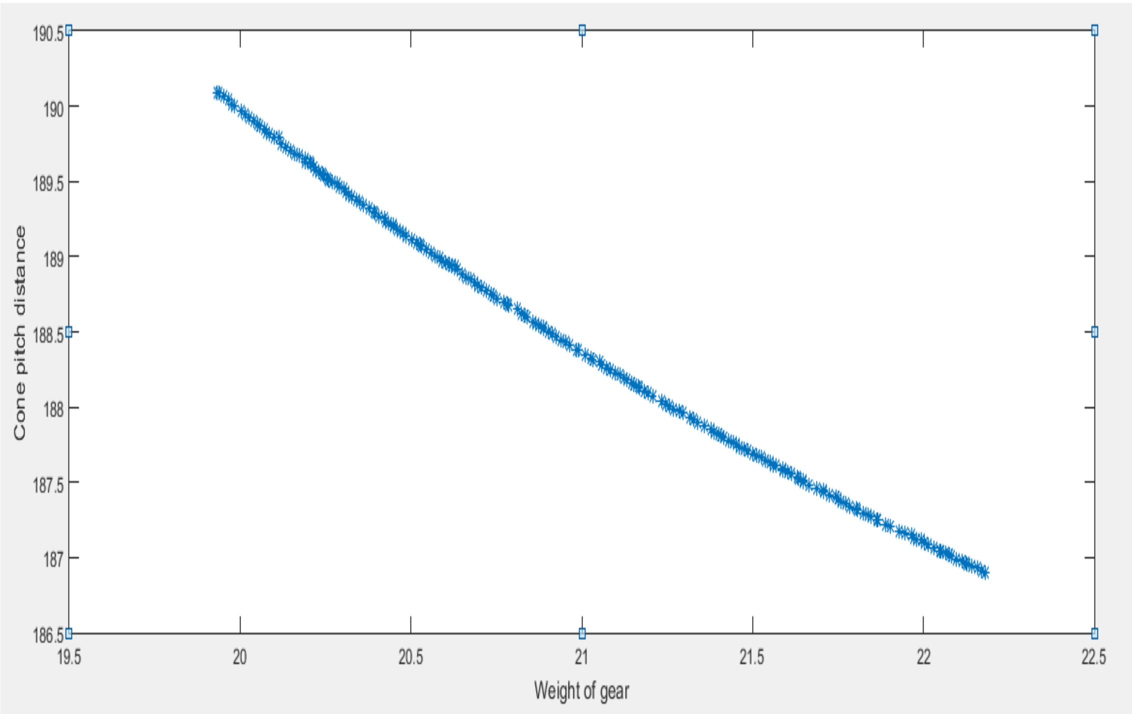

Figure 2. Optimized results (Weight of Gear and Cone distance) by NSGA-II (Case 1)

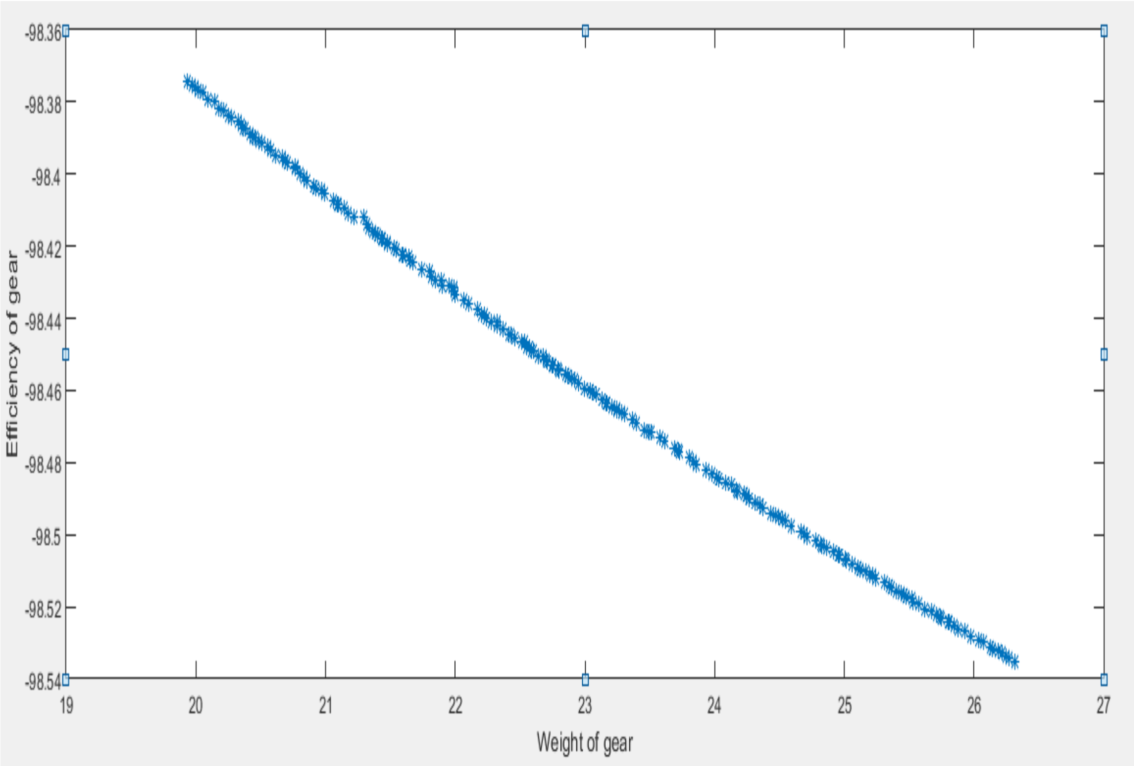

Figure 3. Optimized results (Weight of gear and Efficiency of gear) by NSGA-II (Case 2)

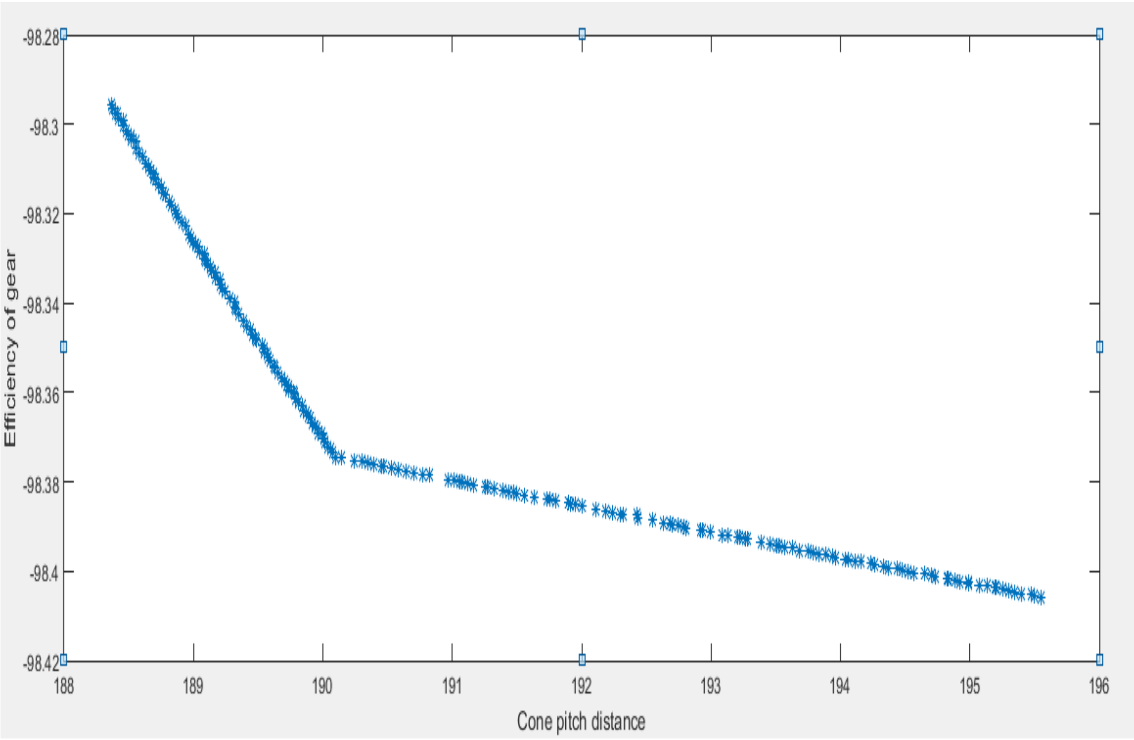

Figure 4. Optimized results (Cone pitch distance and Efficiency of gear) by NSGA-II (Case 3)

In Table 2, the third objective function that was kept as a constraint in each case is shown in bold values. It is also noted that, there is an increase of32.10 % in weight with the additional constraints and design variables as compared to literature. But an increase in efficiency 1.01% is realised than literature results. As cone distance is made as a constraint in Case 2 (the limiting value of the same is the cone distance value of the literature), there is no percentage change. The plots of optimization results are presented Figure 2, Figure 3, Figure 4 and comparison of the same are given in Figure 5.

From Table 3, and Figure 6, it is also noted that design variables, viz. shaft diameter and face width of the gears alone have responded with respect to the different power inputs. But surprisingly there is no such response seen in any of the other design variables irrespective of change in input power.

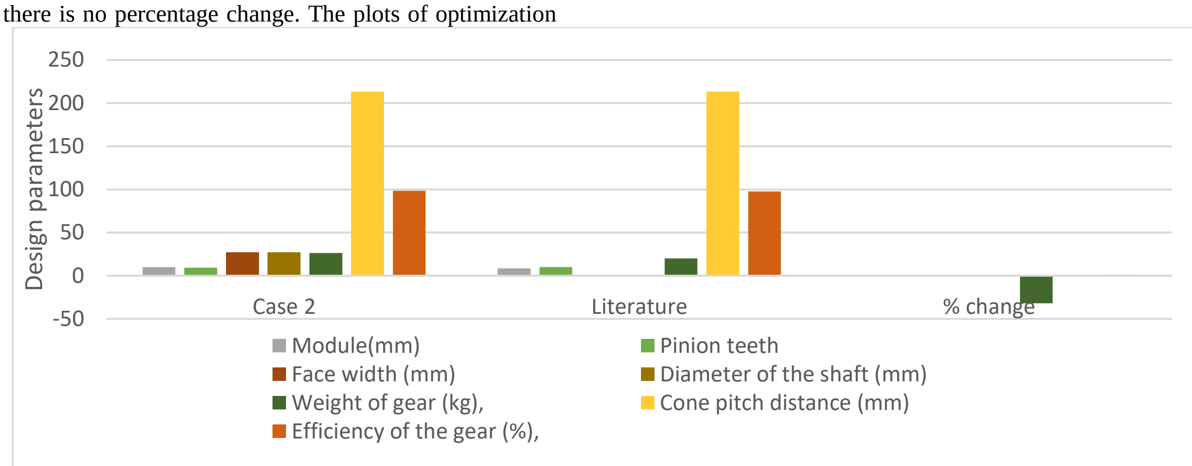

Figure 5. Comparison of results of this work (Case 2) and literature [10]

Table 3. Optimal values (Case 1, Case 2 and Case 3) by NSGA-II of the present research for various power inputs

Parameters	Power 29.828kW			Power 52.199kW		
	Case 1	Case 2	Case 3	Case 1	Case 2	Case 3
Module (mm), m	9.838	9.853	9.356	9.837	9.853	9.356
Pinion teeth , z_1	7.785	9.192	8.565	7.785	9.192	8.565
Face width (mm) b	36.594	33.625	42.398	48.283	33.737	50.174
Diameter of the shaft (mm) , d_s	23.608	36.341	27.370	35.768	35.173	36.597
Weight of gear (kg), W	22.189	26.318	19.923	22.188	26.319	19.923
Cone pitch distance (mm), R_c	186.890	213.625	195.552	186.891	213.625	195.548
Efficiency of the gear (%), $\boldsymbol{\eta}$	97.546	98.535	98.405	97.546	98.535	98.405

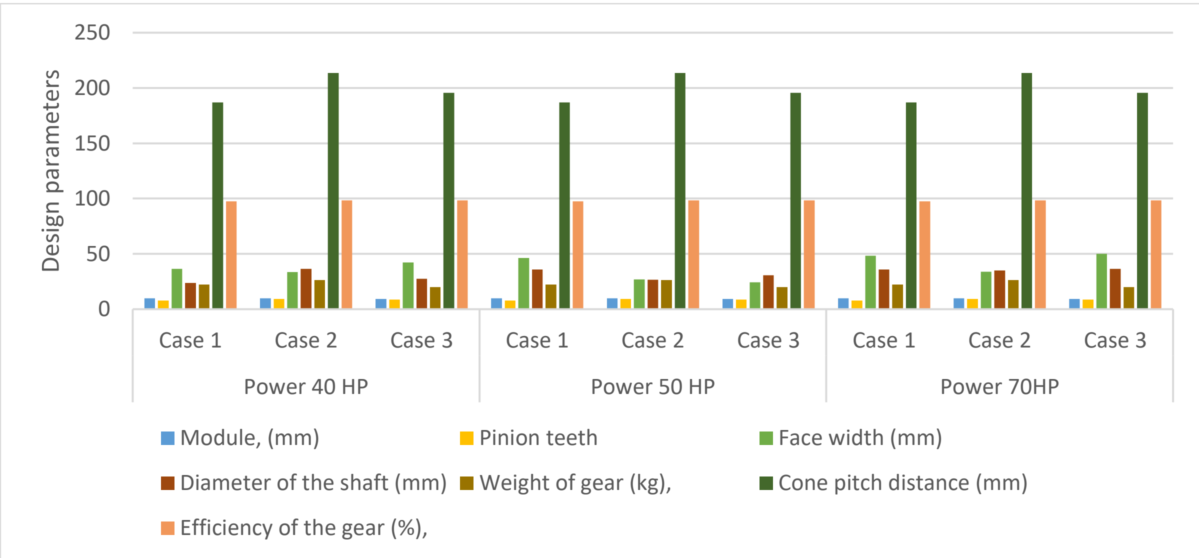

Figure 6. Comparison of results of this work for various power inputs

5. Conclusion and future scope

In this research, optimum parameters for a spiral bevel gear pair with three objective functions along with added design variables and critical mechanical constraints are obtained. The main findings of the research are as follows:

1. The weight of gear has considerably increased from 19.923 kg (for z_1=10) to 22.81 kg (Case 1). It is the same thing in module also, that has increased from 8.729 mm (for z_1=10) to 9.853mm (Case 2) in comparison to literature. It is tolerable as this value has been realised with added critical constraints and deign variables.
2. The finest optimum design parameters (Case 2) are as follows: $= 9.853mm,\ z_1 = 9.192, b = 27.064mm,\ d_s = 23.84\ mm$, and $W = 26.319$ kg, $R_c = 213.625$ mm, and $\eta = 98.535\ \%$.
3. A decrease in cone pitch diameter (186.904 mm) is noted (213.625 mm) for z_1=10than that of literature. In the same way, there is also an increase in efficiency from 97.546% (forz_1=10) to 98.406% (Case 2).It is noteworthy, as these have been accomplished, with added critical constraints and deign variables.
4. There is a considerable rise of 32.10 % weight with additional constraints and design variables than that of the literature. But a surge in efficiency of 1.01% is also observed than that of the literature results.
5. The change in input power had caused design variables, viz. shaft diameter and face width of the gears to respond well. But remarkably there is no such reaction in any of the other design variables. This work is readily applicable and appropriate for optimization of related gear drives used in industries. Certain additional constraints such as profile shift constraint discussed in [24] and tribological constraints can also be considered for future work.

Conflict of interest

On behalf of all authors, the corresponding author states that there is no conflict of interest.

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