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CALCULATING OPTIMAL GEAR RATIOS FOR THREE STAGE BEVEL HELICAL REDUCER FOR MINIMAL REDUCER CROSS-SECTION

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ABSTRACT

This work deals with the determination of optimal gear ratios of three-step bevel helical reducers. To find the optimal ratios, an optimization problem was created and solved. In this problem, the minimal reducer cross section was selected for the target. Also, seven main design parameters counting the total ratio, the allowable contact stresses and the face width coefficients of all steps and the output torque were chosen for the examination of the effects of them to the optimal gear ratios. In addition, to estimate the weight of these parameters on the optimum ratios, a simulation experiment was carried out by programming. From the experimental results, the influence of the design factors on the optimal gear ratios was found, and an equation to find the optimal gear ratios was presented.

Keywords: reducer, gear ratio, optimum gear ratio, three-step bevel helical reducer.

1. INTRODUCTION

In optimization design of a reducer, the calculation of the optimum gear ratios is very important. This is because the mass, the dimension, and the reducer cost are subject strongly to the gear ratios of each step. Therefore, the determination of the optimal gear ratios has attracted the attention of many scientists.

Until now, the optimal gear ratios have been calculated for mechanical systems with different reducer types counting the helical reducers [1-9], the bevel reducers [5, 6, 10-13] or the worm reducers [6, 14-17]. The gear ratios were computed for two-step reducers [2, 3, 5, 10, 11, 18], three-step reducers [1, 7, 9, 12, 19, 20] and four-step reducers [21-24]. Besides, the optimal gear ratios were determined for getting several targets such as minimal length of reducers [1, 14, 20-22, 25], minimal weight of gears [20, 26], the reducer height [27], minimal reducer cost [9], or minimal cross section of reducers [2-4, 19, 20, 23]. The optimal gear ratios have also found for mechanical systems with a reducer and a V-belt drive [28-31]or a chain drive [27, 32-35].

This paper deals with the influence of main design factors on the optimal gear ratios of a three step bevel helical reducer. Also, regression models to predict the optimal gear ratios to get the minimal cross reducer section were suggested.

2. METHODOLOGY

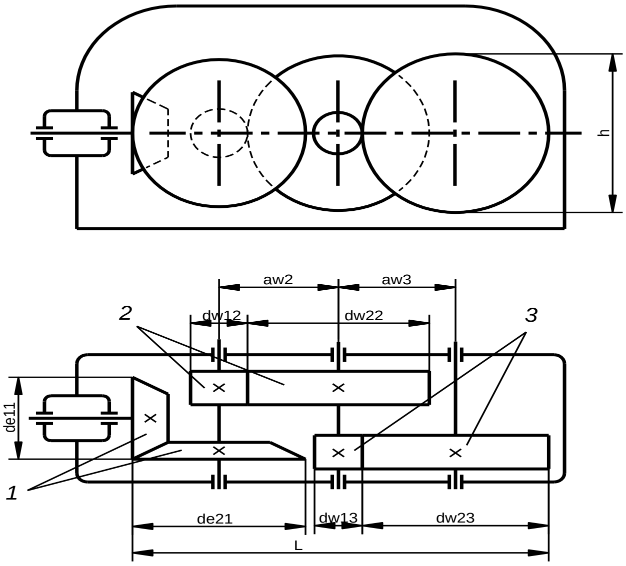


Figure-1. Schema of three-step bevel helical reducer.

For a three-step bevel helical reducer, the reducer cross section can be determined by:

$$A = L \cdot h \tag{1}$$

Wherein L and h are the length and the height of the reducer which are calculated by (see Figure-1):

$$L = d_{e21}/2 + a_{w2} + a_{w3} + d_{w23}/2 \tag{2}$$

$$h = \max\left(d_{e21}, d_{w22}, d_{w23}\right) \tag{3}$$

In Equations (2) and (3), a_{w2}, a_{w3}, d_{w22} and d_{w23} are the center distances and the pitch diameters of steps 2 and 3, respectively. d_{w22} and d_{w23} can be found as [25]:

$$d_{w22} = 2 \cdot a_{w2} \cdot u_2 / (u_2 + 1) \tag{4}$$

$$d_{w23} = 2 \cdot a_{w3} \cdot u_3 / (u_3 + 1) \tag{5}$$

In which, u_2 and u_3 are the gear ratios steps 2 and 3.

From the above analysis, the problem to find the optimum gear ratios is given as:

$$\text{minimize } A = L \cdot h \tag{6}$$

With the constraints as

$$\begin{aligned} &1 \le u_2 \le 9 \\ &1 \le u_3 \le 9 \end{aligned} \tag{7}$$

From equations (1), (2), (4), (5) and (6), to solve the above problem, it is essential to define d_{e21}, a_{w2} and a_{w3}.

2.1 Calculating d_{e21}

For the bevel gear set, d_{e21} can be found as [25]:

$$d_{e21} = 2 \cdot u_1 \cdot R_e / \sqrt{1 + u_1^2} \tag{8}$$

Where, R_e is the distance of external cone; R_e is calculated by [25]:

$$R_e = k_R \cdot \sqrt{u_1^2 + 1} \cdot \sqrt[3]{T_{11} \cdot k_{H\beta 1} / \left[(1 - k_{be}) \cdot k_{be} \cdot u_1 \cdot [\sigma_H]^2 \right]} \tag{9}$$

In which, $k_R = 50$ (MPa1/3) is material coefficient; u_1 is the gear ratio of step 1; $k_{be} = 0.25 \ldots 0.3$ is the face width coefficient; $[\sigma_{H1}]$ is the allowable contact stress (MPa); $K_{H\beta 1}$ is the concentration factor of load; From the data in [25], $K_{H\beta 1}$ is determined by:

$$K_{H\beta 1} = 0.25 \cdot k^2 + 0.2 \cdot k + 1.02 \tag{10}$$

Wherein, $k = k_{be} \cdot u_1 / (2 - k_{be})$. For the reducer, the following equation can be written:

$$T_{11} = T_{out} / \left(u_g \cdot \eta_{bg} \cdot \eta_{hg} \cdot \eta_b^3 \right) \tag{11}$$

In which, T_{out} is the output torque (Nmm); u_g is the total ratio of the reducer; $\eta_{hg} = 0.95 \ldots 0.97$ is the bevel gear efficiency [25]; $\eta_{hg} = 0.96 \ldots 0.98$ is the helical gear efficiency [25]; $\eta_b = 0.99 \ldots 0.995$ is the bearing efficiency [25]. Choosing $\eta_{bg} = 0.96$, $\eta_{hg} = 0.97$, $\eta_b = 0.992$ and replacing them into (11) provides:

$$T_{11} = 1.101 \cdot T_{out} / u_g \tag{12}$$

Replacing $k_R = 50$ and (13) into (10) gets

$$R_e = 51.6296 \cdot \sqrt{u_1^2 + 1} \cdot \sqrt[3]{T_{out} \cdot k_{H\beta 1} / \left[(1 - k_{be}) \cdot k_{be} \cdot u_1 \cdot u_g \cdot [\sigma_H]^2 \right]} \tag{13}$$

2.2 Calculating a_{w2}

For helical gear set, a_{w2} can be determined by [25]:

$$a_{w2} = k_m \cdot (u_2 + 1) \cdot \sqrt[3]{\frac{T_{12} \cdot k_{H\beta}}{[\sigma_H]^2 \cdot u_2 \cdot \psi_{ba2}}} \tag{14}$$

In which, $K_{H\beta}$ is the concentration factor of load; it can be chosen by $k_{H\beta} = 1.1$ as $k_{H\beta} = 1.02 \div 1.28$ [25]; $[\sigma_H]$ is the allowable contact stress (MPa); k_m =43 is the material coefficient [25]; ψ_{ba2} is the wheel face width coefficient of step 2.

For the three step bevel helical reducer, the output torque can be calculated by:

$$T_{out} = T_{12} \cdot \eta_{hg}^2 \cdot \eta_b^3 \cdot u_2 \cdot u_3 \tag{15}$$

Replacing $\eta_{hg} = 0.97$ and $\eta_b = 0.992$ as in sub-section 2.1 have

$$T_{12} = 1.0887 \cdot T_{out} / (u_2 \cdot u_3) \tag{16}$$

Substituting (16) and $k_{H\beta} = 1.1$ into equation (14) with considering $u_2 \cdot u_3 = u_g / u_1$ have:

$$a_{w2} = 45.6635 \cdot (u_2 + 1) \cdot \sqrt[3]{\frac{T_{out} \cdot u_1}{[\sigma_H]^2 \cdot u_g \cdot u_2 \cdot \psi_{ba2}}} \tag{17}$$

2.3 Calculating a_{w3}

For the third step, a_{w3} is calculated by [25]:

$$a_{w3} = K_m \cdot (u_3 + 1) \cdot \sqrt[3]{\frac{T_{13} \cdot k_{H\beta}}{[\sigma_H]^2 \cdot u_3 \cdot \psi_{ba3}}} \tag{18}$$

Also

$$T_{out} = T_{13} \cdot \eta_{hg} \cdot \eta_b^2 \cdot u_3 \tag{19}$$

Selecting $\eta_{hg} = 0.97$ and $\eta_b = 0.992$ as in section 2.2 have

$$T_{12} = 1.0476 \cdot T_{out} / u_3 \tag{20}$$

Substituting (20), k_m =43 and $k_{H\beta} = 1.1$ (as in section 2.2) into (18) gets:

$$a_{w3} = 45.0814 \cdot (u_3 + 1) \cdot \sqrt[3]{\frac{T_{out}}{[\sigma_H]^2 \cdot u_3^2 \cdot \psi_{ba3}}} \tag{21}$$

2.4 Experimental work

In order to discover the impact of the design factors on the optimum ratios, a simulation experiment was planned and accomplished. For this work, the factorial design was taken and a 2 level factorial design with ½ fraction was carefully chosen. In addition, 8 main design factors were selected for the analysis (Table-1). Consequently, the experiment was organised with $2^{8-1} = 128$ test runs. Also, based on equations (6) and (7), a computational program was built for solving the problem. The plan of the experiment and the optimum ratios u_1 and u_2 are shown in Table-2.

Table-1. Input parameters.

Factor	Code	Unit	Low	High
Total reducer ratio	u_g	-	40	110
Face width coefficient of the bevel gear set	Kbe	-	0.25	0.3
Wheel face width coefficient of step 2	x_{ba2}	-	0.3	0.35
Wheel face width coefficient of step 3	x_{ba3}	-	0.35	0.4
Allowable contact stress of step 1	AS_1	MPa	360	420
Allowable contact stress of step 2	AS_2	MPa	360	420
Allowable contact stress of step 3	AS_3	MPa	360	420
Output torque	T_{out}	Nmm	10^5	10^7

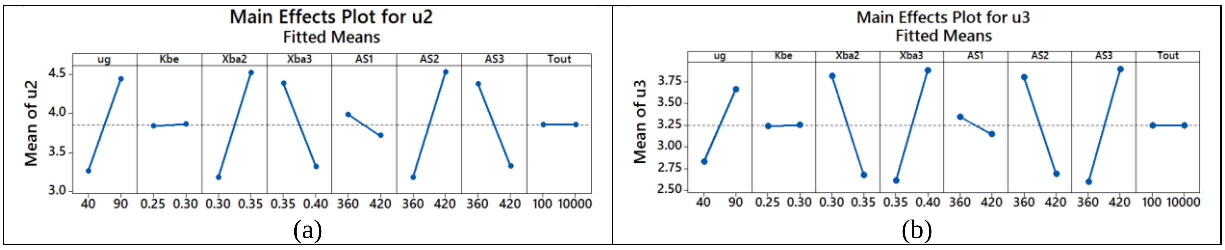
Table-2. Plan of experiment and output response.

Std Order	Run Order	Center Pt	Blocks	u_g	K_{be}	Xba2	Xba3	AS_1 (MPa)	AS_2 (MPa)	AS_3 (MPa)	Tout (Nm)	u_2	u_3
26	1	1	1	90	0.25	0.3	0.4	420	360	360	10000	1.50	5.64
46	2	1	1	90	0.25	0.35	0.4	360	420	360	100	6.03	2.31
110	3	1	1	90	0.25	0.35	0.4	360	420	420	10000	6.60	2.82
8	4	1	1	90	0.3	0.35	0.35	360	360	360	10000	5.70	2.25
113	5	1	1	40	0.25	0.3	0.35	420	420	420	10000	3.99	2.25
98	6	1	1	90	0.25	0.3	0.35	360	420	420	10000	6.15	2.79
…													
122	127	1	1	90	0.25	0.3	0.4	420	420	420	10000	1.50	6.12
119	128	1	1	40	0.3	0.35	0.35	420	420	420	10000	3.96	1.98

3. RESULTS AND DISCUSSIONS

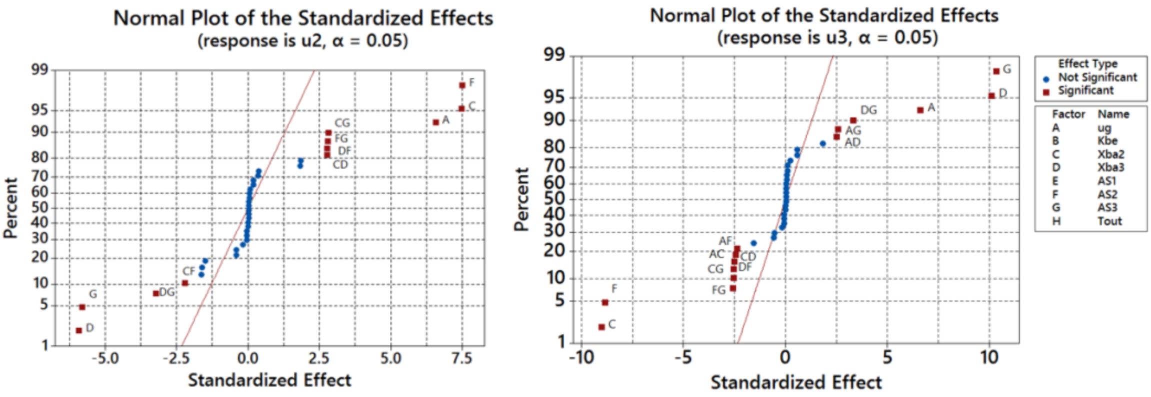
Figure-2 shows the impact of the main design factors on the optimal ratios of the second step u_2 (Figure-2a) and the third step u_3 (Figure-2b). From the Fig., u_2 increases significantly with the increase of u_g, ψ_{ba2} and AS_2. On the other hand, u_3 upsurges considerably with the growth of u_g, ψ_{ba3} and AS_3. In addition, with the rise of ψ_{ba2} and AS$_2$, u_3 declines remarkably, whereas, with the increase of ψ_{ba3} and AS$_3$, u_2 reduces significantly. Besides, both u_2 and u_3 decrease with the increase of the AS$_1$. Additionally, they are nearly not affected by k_{be} and T_{out}.

Figure-3 presents the Normal Plot of the standardized effects for u_2 (Figure-3.a) and u_3 (Figure-3b). It is found from the figure that u_g (factor A), ψ_{ba2} (factor C), ψ_{ba3} (factor D), AS$_2$ (factor F) and AS$_3$ (factor G) are the most weighty factors for both u_2 and u_3. Furthermore, u_g (factor A), ψ_{ba2} (factor C), AS$_2$ (factor F) and the interactions CG, FG, DF and CD have a positive impact for u_2. Besides, ψ_{ba3} (factor D), AS$_3$ (factor G) and the interactions DG and CF have a negative impact for u_2. On the other hand, for u_3, factors A, D, G and the interactions AD, AG and DG a positive standardized effect and factors C, F and the interactions AC, AF, CG and FG have a negative influence.



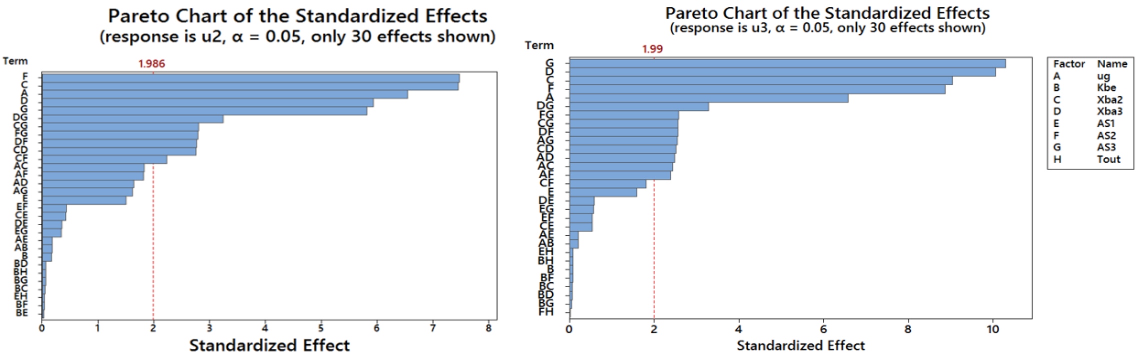
(a) (b)

Figure-2. Graph of main influences for u_2 and u_3.



a) b)

Figure-3. Normal plot for u_2 and u_3.



a) b)

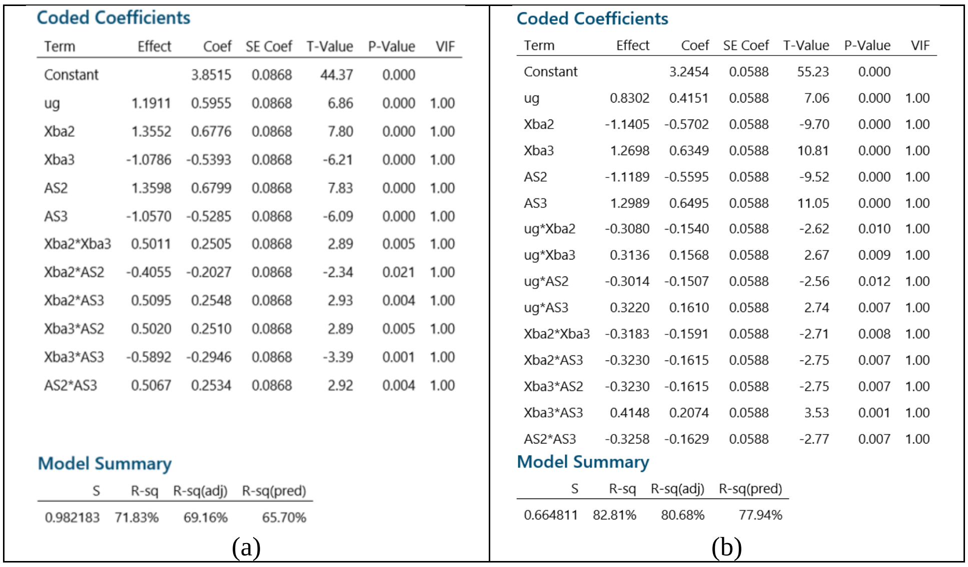
Figure-4. Pareto chart for u_2 and u_3.

The Pareto chart for u_2 (Figure-4a) and u_3 (Figure-4b) is described in Figure-4. It is learned that the reference line crosses the bars which represent factors F, C, A, D, G and the interactions DG, CG, FG, DF and CD. As a result, these factors are significant with the optimum gear ratios of the second step u_2. Also, the bars which characterize factors G, D, C, F, A and the interactions DG, FG, CG, DF, AG, CD, AD, AC cut the reference line. Consequently, these parameters are weighty with u_3.

Figure-5 shows the expected effects and coefficients for u_2 (Figure-5a) and u_3 (Figure-5b). From this Fig., parameters which have P-values lesser than 0.05 are u_g, ψ_{ba2} and ψ_{ba3}, AS$_2$ and AS$_3$, and the interactions $\psi_{ba2} * \psi_{ba3}$, $\psi_{ba2} * AS_2$, $\psi_{ba2} * AS_3$, $\psi_{ba3} * AS_2$, $\psi_{ba3} * AS_3$ and $AS_2 * AS_3$. Subsequently, these parameters have a significant effect on the responses (u_2 and u_3). Also, the following models are proposed for finding u_2 and u_3:

$$u_2 = 91.6 + 0.02382 \cdot u_g - 150.3 \cdot \psi_{ba2} - 129.2 \cdot \psi_{ba3} - 0.1248 \cdot AS_2 - \\ -0.0905 \cdot AS_3 + 401 \cdot \psi_{ba2} \cdot \psi_{ba3} - 0.27 \cdot \psi_{ba2} \cdot AS_2 + 0.34 \cdot \psi_{ba2} \cdot AS_3 + \\ +0.335 \cdot \psi_{ba3} \cdot AS_2 - 0.393 \cdot \psi_{ba3} \cdot AS_3 + 0.000282 \cdot AS_2 \cdot AS_3 \qquad (22)$$

$$u_3 = -76.7 - 0.0028 \cdot u_g + 172.7 \cdot \psi_{ba2} + 68 \cdot \psi_{ba3} + 0.1457 \cdot AS_2 + \\ +0.0445 \cdot AS_3 - 0.2464 \cdot u_g \cdot \psi_{ba2} + 0.2509 \cdot u_g \cdot \psi_{ba3} - 0.000201 \cdot u_g \cdot AS_2 + \\ +0.000215 \cdot u_g \cdot AS_3 - 254.6 \cdot \psi_{ba2} \cdot \psi_{ba3} - 0.2153 \cdot \psi_{ba2} \cdot AS_3 - \\ -0.2153 \cdot \psi_{ba3} \cdot AS_2 + 0.2766 \cdot \psi_{ba3} \cdot AS_3 - 0.000181 \cdot AS_2 \cdot AS_3 \qquad (23)$$



Coded Coefficients

Term	Effect	Coef	SE Coef	T-Value	P-Value	VIF
Constant		3.8515	0.0868	44.37	0.000	
ug	1.1911	0.5955	0.0868	6.86	0.000	1.00
Xba2	1.3552	0.6776	0.0868	7.80	0.000	1.00
Xba3	-1.0786	-0.5393	0.0868	-6.21	0.000	1.00
AS2	1.3598	0.6799	0.0868	7.83	0.000	1.00
AS3	-1.0570	-0.5285	0.0868	-6.09	0.000	1.00
Xba2*Xba3	0.5011	0.2505	0.0868	2.89	0.005	1.00
Xba2*AS2	-0.4055	-0.2027	0.0868	-2.34	0.021	1.00
Xba2*AS3	0.5095	0.2548	0.0868	2.93	0.004	1.00
Xba3*AS2	0.5020	0.2510	0.0868	2.89	0.005	1.00
Xba3*AS3	-0.5892	-0.2946	0.0868	-3.39	0.001	1.00
AS2*AS3	0.5067	0.2534	0.0868	2.92	0.004	1.00

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.982183	71.83%	69.16%	65.70%

(a)

Coded Coefficients

Term	Effect	Coef	SE Coef	T-Value	P-Value	VIF
Constant		3.2454	0.0588	55.23	0.000	
ug	0.8302	0.4151	0.0588	7.06	0.000	1.00
Xba2	-1.1405	-0.5702	0.0588	-9.70	0.000	1.00
Xba3	1.2698	0.6349	0.0588	10.81	0.000	1.00
AS2	-1.1189	-0.5595	0.0588	-9.52	0.000	1.00
AS3	1.2989	0.6495	0.0588	11.05	0.000	1.00
ug*Xba2	-0.3080	-0.1540	0.0588	-2.62	0.010	1.00
ug*Xba3	0.3136	0.1568	0.0588	2.67	0.009	1.00
ug*AS2	-0.3014	-0.1507	0.0588	-2.56	0.012	1.00
ug*AS3	0.3220	0.1610	0.0588	2.74	0.007	1.00
Xba2*Xba3	-0.3183	-0.1591	0.0588	-2.71	0.008	1.00
Xba2*AS3	-0.3230	-0.1615	0.0588	-2.75	0.007	1.00
Xba3*AS2	-0.3230	-0.1615	0.0588	-2.75	0.007	1.00
Xba3*AS3	0.4148	0.2074	0.0588	3.53	0.001	1.00
AS2*AS3	-0.3258	-0.1629	0.0588	-2.77	0.007	1.00

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.664811	82.81%	80.68%	77.94%

(b)

Figure-5. Estimated effects and coefficients for u_2 and u_3.

4. CONCLUSIONS

The present work deals with the determination of the optimum gear ratios of a three step bevel helical reducer to find the minimal reducer cross section was presented. In this work, the effect of the main design factors excluding the entire reducer ratio, the wheel face width coefficients and the allowable contact stresses of all steps, and the output torque were inspected. Also, regression models to determine the optimal gear ratios of each step of the reducer were suggested. The optimal ratios of the reducer can be calculated simply as they are explicit.

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REFERENCES

[1] Pi V.N. 2008. A new study on optimal calculation of partial transmission ratio of three-step helical reducers. in The 3rd IASME/WSEAS international conference on continuum mechanics, Cambridge, UK.

[2] Ngoc Pi, V. 2001. A method for optimal calculation of total transmission ratio of two step helical gearboxes. in Proceedings of the National Conference on Engineering Mechanics.

[3] Tuan N.K., *et al*. 2018. Determining optimal gear ratios of a two-stage helical reducer for getting minimal acreage of cross section. in MATEC Web of Conferences. EDP Sciences.

[4] Pi V.N. and N.K. Tuan. 2016. Optimum determination of partial transmission ratios of three-step helical gearboxes for getting minimum cross section dimension. Journal of Environmental Science and Engineering A. 5: 570-573.

[5] Kudreavtev V., I. Gierzaves and E. Glukharev. 1971. Design and calculus of gearboxes. Mashinostroenie Publishing, Sankt Petersburg.

[6] Milou G., Dobre G., Visa F., Vitila H. 1996. Optimal design of two step gear units, regarding the main parameters. VDI Berichte. (1230): 227-235.

[7] Pi V.N., *et al*. A study on calculating optimum gear ratios of a three-stage helical gearbox.

[8] Pi V.N. 2008. A New Study on Optimal Calculation of Partial Transmission Ratios of Helical Gearboxes with First-step Double Gear-sets. in Proceedings of the International Multi Conference of Engineers and Computer Scientists.

[9] Vu N.-P., *et al*. 2020. The Influence of Main Design Parameters on the Overall Cost of a Gearbox. Applied Sciences 10(7): 2365.

[10] Ngoc Pi, V. 2000. A new and effective method for optimal calculation of total transmission ratio of two step bevel-helical gearboxes. in Proceedings of the International Colloquium on Mechannics of Solids, Fluids, Structures & Interaction.

[11] 1Hong T.T., *et al*. 2019. Calculating optimum gear ratios of two step bevel helical reducer. Int. J. Appl. Eng. Res 14: 3494-3499.

[12] Pi V.N., *et al*. 2019. Determination of optimum gear ratios of a three stage bevel helical gearbox. in IOP Conference Series: Materials Science and Engineering. IOP Publishing.

[13] Hong T.T., *et al*. 2006. A study on calculation of optimum gear ratios for three stage bevel helical gearboxes. a a. 500: 1.

[14] Pi V.N. and V.Q. Dac. 2007. Optimal calculation of partial transmission ratios of worm-helical gear reducers for minimal gearbox length. J. Sci. Technol. Tech. Univ. 61: 73-77.

[15] Pi V.N. and V.Q. Dac. 2007. Calculation of total transmission ratio of two step worm reducers for the best reasonable gearbox housing structure. J. Sci. Technol. Thai Nguyen Univ. 1(41): 65-69.

[16] Pi V.N. and V.Q. Dac. 2005. Optimal calculation of total transmission ratio of worm-helical gear reducers. J. Sci. Technol. Thai Nguyen Univ. 4(36): 70-73.

[17] Pi V.N., *et al*. 2019. A study on determination of optimum gear ratios of a worm-helical gearbox. in IOP Conference Series: Materials Science and Engineering. IOP Publishing.

[18] Pi V.N., *et al*. 2019. Determination of optimum gear ratios of a two-stage helical gearbox with second stage double gear sets. in IOP Conference Series: Materials Science and Engineering. IOP Publishing.

[19] Pi V.N. 2008. A new study on optimal calculation of partial transmission ratios of three-step helical reducers for getting minimal cross section dimension. in The 2nd WSEAS International Conference on Computer Engineering and Applications (CEA'08), Acapulco, Mexico.

[20] Pi V.N. 2008. Optimum determination of partial transmission ratios of three-step helical gearboxes. WSEAS Transactions on Applied and Theoretical Mechanics. 3: 43-42.

[21] Pi V.N. 2008. Optimal calculation of partial transmission ratios for four-step helical gearboxes with first and third-step double gear-sets for minimal gearbox length. in American Conference on Applied Mathematics (MATH 2008), Harvard, Massachusetts, USA.

[22] Pi V.N. 2008. Optimal calculation of partial transmission ratios of four-step helical gearboxes for getting minimal gearbox length. World Academy of Science, Engineering and Technology.

[23] Pi V.N. 2008. Optimal Calculation of Partial Transmission Ratios of Four-step Helical Gearboxes for Getting Minimal Cross Section Dimension. in Proceedings of the International Multi Conference of Engineers and Computer Scientists.

[24] Pi V. 2008. A study on optimal calculation of partial transmission ratios of four-step helical gearboxes with second and fourth step double gear-sets. in WSEAS International Conference. Proceedings. Mathematics and Computers in Science and Engineering. World Scientific and Engineering Academy and Society.

[25] Pi V.N. 2008. Optimal determination of partial transmission ratios of three-step helical gearboxes with second-step double gear-sets for minimal

gearbox length. in Proceedings of the 2nd WSEAS International Conference on Computer Engineering and Applications. World Scientific and Engineering Academy and Society (WSEAS).

[26] Pi V.N. 2008. Optimal determination of partial transmission ratios of three-step helical gearboxes with first and third-step double gear-sets for minimal mass of gear. in American Conference on Applied Mathematics, MATH.

[27] Cam N.T.H., *et al*. 2018. A study on determination of optimum partial transmission ratios of mechanical driven systems using a chain drive and a three-step helical reducer. in International Conference on Engineering Research and Applications. 2018. Springer.

[28] Tuan N.K., *et al*. 2018. Optimum calculation of partial transmission ratios of mechanical driven systems using a V-belt and a three-step bevel helical gearbox. in International Conference on Engineering Research and Applications. Springer.

[29] Pi V.N., T.T.P. Thao and L. Thao. 2015. A new study on optimum determination of partial transmission ratios of mechanical driven systems using a V-belt and two-step helical gearbox. Vietnam Mechanical Engineering Journal. 10: 123-5.

[30] Tuan D.A., N.K. Tuan and V.N. Pi. 2017. Optimum determination of partial transmission ratios of mechanical driven systems using a V-belt and a three-step helical gearbox. J. Environ. Sci. Eng. B. 6: 328-331.

[31] Tuan N.K. and V.N. Pi. 2017. Optimum Determination of Partial Transmission Ratios of Mechanical Driven Systems Using a V-belt and a Helical Gearbox with Second-Step Double Gear-Sets. Journal of Environmental Science and Engineering A. p. 377.

[32] Hung L.X., *et al*. 2019. Calculation of Optimum Gear Ratios of Mechanical Driven Systems Using Two-Stage Helical Gearbox with First Stage Double Gear Sets and Chain Drive. in International Conference on Engineering Research and Applications. Springer.

[33] Thao T.T.P., *et al*. 2019. Determining Optimum Gear Ratios of Mechanical Driven Systems Using Three Stage Bevel Helical Gearbox and Chain Drive. in International Conference on Engineering Research and Applications. Springer.

[34] Tuan N.K., *et al*. 2019. A Study on Determining Optimum Gear Ratios of Mechanical Driven Systems Using Two-Step Helical Gearbox with First Step Double Gear Sets and Chain Drive. in International Conference on Engineering Research and Applications. Springer.

[35] Pi V.N., N.K. Tuan and L.X. Hung. 2020. A New Study on Calculation of Optimum Partial Transmission Ratios of Mechanical Driven Systems Using a Chain Drive and a Two-Stage Helical Reducer, in Advances in Material Sciences and Engineering. Springer. pp. 97-10.