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Kinematic Synthesis of a Crossed Four-bar Mechanism for Automotive Steering

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ABSTRACT:

Kinematic synthesis has been carried out for crossed four-bar mechanisms to be used in steering mechanism of four wheel vehicles with several values of track to wheelbase ratios. Two equal spur gears have been included in the mechanism. Initial estimate has been made for the link size and angular orientation for straight ahead motion using iterative techniques. Hooke and Jeeves optimization method has been used to carry out the final estimation of the two design parameters. The objective function comprises of the steering errors only. Five precision points have been found out in the entire range of rotation of the front wheels. A very low steering error has been achieved by the kinematic synthesis. The limitation of this mechanism has been discussed and suitable modification has been suggested.

KEYWORDS:

Kinematic synthesis; Steering mechanism; Crossed four-bar linkage; Hooke and Jeeves optimization

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1. Introduction

Kinematic synthesis methods can be broadly classified into two categories: exact point approach and optimization techniques. In the exact point approach there are some precision points at which the function generated is exactly same as the required function. At other points there are deviations from the required function. This deviation is called the structural error. In order to reduce the structural error the number of precision points has to be increased. Generally the number of precision points is equal to the number of design parameters. The design parameters are the link length ratios and the angles used to define a mechanism. A four-bar steering mechanism is generally used in buses and trucks. The mechanism is placed behind the front axle. This mechanism is called Ackermann steering. This mechanism is simple but has a drawback that there is divergent end behavior in the steering error curve. Fahey and Huston [1] modified a four-bar mechanism to an eight-bar mechanism in order to remove the divergent end behavior in an extended range of motion.

They used an iterative method to achieve the final solution from the initial solution. But one link of the mechanism was small compared to other links. The wear in this small link joint may affect the accuracy of the mechanism greatly. The eight-bar mechanism provided seven precision points but is very complex. Pramanik [2] considered a six-bar mechanism for steering of an automobile. This mechanism provided five precision points and low steering error. He used algebraic method to find the three design parameters with the application of Newton-Raphson method. De-Juan Ana et al [3] carried out optimal synthesis of function generation in steering linkages. They considered rack-and-pinion steering, four-bar steering, six-bar steering and double four-bar steering mechanism. In all these cases they considered leading configuration, trailing configuration, mixed-leading configuration and mixed-trailing configurations. However, they did not consider the mixed leading-and-trailing configuration of the four-bar mechanism. The present work is concerned about this mixed configuration and design of a new mechanism.

Jing-Shan Zhao et al [4] considered a planar five bar mechanism with two incomplete noncircular gears for steering of an automobile. The mechanism is able to follow the Ackermann equation exactly because the gears are designed in such a way. But the mechanism is mainly suitable for light carriages. M.M. Ettefagh and M.S. Javash [5] carried out optimal synthesis of four-bar steering mechanism using AIS and genetic algorithms. In the first method lengths were selected as optimization parameters whereas in the second method the precision point distributions were considered as optimization parameters. It has been found that the second method produced infinitesimal error but could not satisfy the dimensional constraints.

2. Crossed four-bar mechanism

A crossed four-bar mechanism AGFB has been shown in the Fig. 1. In this mechanism the links AG and BF are equal in size which is one design parameter. These links are inclined to the longitudinal axis of the vehicle by equal angle β when the vehicle moves along a straight path. The angles GAK and FBH are equal to β which is another design parameter. When the outer wheel rotates by angle GAD then the inner wheel rotates by angle FBE. The angle DAB is designated by θ. The extreme precision position of the mechanism is shown by ADEB. The lines AD and EB are extended to intersect at point C. The angle CBA is designated by γ. Since the arms AG and BF rotate in the opposite directions two equal spur gears have been used in order to get coordinated motion of the front wheels. The two centers of the spur gears are B and I. The two hinge centers of the outer and inner wheels are A and I respectively.

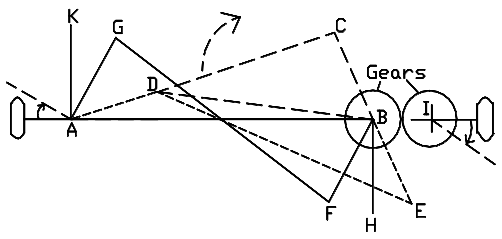

Fig. 1: A crossed four-bar steering mechanism including two gears

3. Initial estimate

It has been assumed that the distance d between hinge joints A and B is 10 units. The arms AG and BF are equal to r. The length of the coupler link GF is given by

$$s = \sqrt{\left\{(d - 2r\sin\beta)^2 + (2r\cos\beta)^2\right\}}$$

The rotation of the inner wheel θ_i is

$$\angle FBE = \left(\frac{\pi}{2} - \gamma + \beta\right)$$

The rotation of the outer wheel θ_o is

$$\angle GAD = \frac{\pi}{2} - \theta - \beta$$

From the sine rule of triangle ABC one gets

$$\frac{AB}{\sin(\pi - \theta - \gamma)} = \frac{CD + r}{\sin\gamma} = \frac{CE - r}{\sin\theta}$$

Now the two sides of the triangle CDE are given by

$$CD = AB \cdot \frac{\sin\gamma}{\sin(\theta + \gamma)} - r \tag{1}$$

$$CE = AB \cdot \frac{\sin\theta}{\sin(\theta + \gamma)} + r \tag{2}$$

If these two sides are equal then the $\angle$CDE and CED are equal. Then the transmission angle of the mechanism at joints D and E will be equal. The precision of the mechanism is more important than the transmission angle at the joints at extreme position. Therefore instead of equating the two sides the following relation has been assumed.

$$CD = n \cdot CE \tag{3}$$

Here n is a real constant with possibility of becoming more than or less than unity. It has been found that if this is little less than unity five precision points are obtained. From the Eqns. (1-3) the equal arms AD and BE are found as,

$$r = \frac{d(\sin\gamma - n\sin\theta)}{(1+n)\sin(\theta + \gamma)} \tag{4}$$

4. Calculation procedure

a) The wheel track to wheelbase ratio has been taken for the range from 0.2 to 0.6.
b) The rotation of the inner wheel has been assumed to be around 70°. The required correct rotation of the outer wheel has been calculated from Ackermann equation.
c) The value of n has been taken 0.9 .To begin the iterations we vary the angle γ from 20° to 60°. The $\angle\theta$ has been calculated from the following relation. $\theta = \pi - \gamma - \theta_i - \theta_o$
d) The length of steering arm r has been calculated from Eqn. (4).
e) We check whether the length of the coupler DE and FG are equal or not. In other words we find that $\angle\gamma$ for which the lengths GF and DE are equal.
f) Initial estimate for the length r has been obtained from Eqn. (4) and $\angle\beta$ has been obtained from geometry.
g) After this steering error curve has been obtained by geometrical calculations.

5. Results

The calculated values for five precision point solutions have been shown in Table 1. It has been found from Table 1 that the length of the steering arm r increases as the track to wheelbase ratio increases. The limitation of this mechanism is that the steering arm length r is very large for higher values of track to wheelbase ratios.

Table 1: Initial estimates of the steering mechanisms

Track to wheelbase ratio	Assumed rotation of inner wheel (°)	Value of n	Length of steering arm r	Maximum steering error (°)	Precision position of inner wheel (°)
0.2	70	0.9	1.97	0.234	60
0.3	70	0.9	3.25	0.563	56
0.4	70	0.8	4.03	0.406	52
0.5	70	0.7	4.67	0.239	48
0.6	80	0.7	4.91	1.03	52

6. Optimization method

The mechanism has two design parameters. These are $\angle$KAG (β) and steering arm length AG (r). The outer wheel has been rotated by $\angle\alpha$ ($\angle$GAD) and then the rotation of the inner wheel ($\angle$FBE) has been found out. The correct rotation of the inner wheel has been found out using the law of correct steering. The initial straight ahead position has two equal $\angle$KAG and FBH. The two steering arms AG and FB are equal. The length of the coupler has been found as,

$$GF = \sqrt{(d - 2r\sin\beta)^2 + (2r\cos\beta)^2}$$

The distance DB is given by

$$DB = \sqrt{\{d - r\sin(\alpha + \beta)\}^2 + \{r\cos(\alpha + \beta)\}^2}$$

The $\angle$DBE is given by

$$\angle DBE = \cos^{-1}\left[\frac{(DB)^2 + r^2 - (GF^2)}{2 \times DB \times r}\right]$$

The distance DF is given by

$$DF = \sqrt{\{d - r\sin\beta - r\sin(\alpha+\beta)\}^2 + \{r\cos\beta + r\cos(\alpha+\beta)\}^2}$$

The $\angle$ DBF is given by

$$\angle DBF = \cos^{-1}\left[\frac{(DB)^2 + r^2 - (DF^2)}{2\times DB\times r}\right]$$

The rotation of the inner wheel is given by

$$\angle FBE = \angle DBE - \angle DBF$$

The correct angle of rotation of the inner wheel is given by

$$\angle FBE_{correct} = \cot^{-1}\left(\cot\alpha - \frac{t}{w}\right)$$

Where t is wheel track and w is wheelbase. The steering error is given by

$$Error = \angle FBE - \angle FBE_{correct}$$

The objective function is given by

$$Obj\,Fun = \sum (Error)^2$$

The Hooke and Jeeves optimization method has been used to minimize the objective function using the initial estimate as the initial solution. A vehicle with track to wheelbase ratio 0.4 has been considered for which the final estimate has been made. The length of the steering arm r has been found as 4.064 units where the distance between hinge joints AB is 10 units. The inclination β of the steering arms with the vehicle longitudinal axis is 28.56°. The steering error curve has been plotted in Fig. 2. The maximum steering error has been found 0.12° and three precision points are noticed in Fig. 2.

Fig. 2: Steering error curve

7. Conclusion

The proposed new steering mechanism is suitable for long wheelbase vehicle particularly for a track to wheelbase ratio 0.4. For track to wheelbase ratio four-tenth five precision points have been obtained in the entire range and the maximum steering error is 0.12°. But the length of steering arm is about 40% of the track width. The steering arm has been reduced to 20% by suitably modifying the mechanism as in Fig. 3.

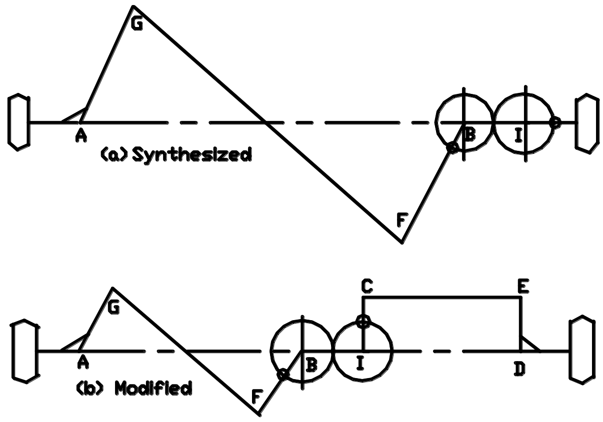

Fig. 3: Modified steering mechanism

In this modification the crossed four bar mechanism has been reduced to half size and the two gears have been brought to the mid-position with introduction of a parallel four bar mechanism. The use of gears at the end is difficult to assemble the steering knuckle. By bringing the gears at the middle this problem is eliminated. The advantage of the suggested mechanism is that steering error is very small with five precision points and the divergent end behavior of Ackermann steering mechanism has been eliminated in an extended range of motion. The length of the steering arm is about twenty percent of the wheel track which is acceptable. This work includes the configuration of crossed four bar mechanism for kinematic synthesis that was not done in [3]. The use of gears may affect the accuracy of the steering if there is backlash in the gear. Hence this steering mechanism is suitable for light duty vehicles.

REFERENCES:

[1] S.O'F. Fahey and D.R .Huston. 1997. A novel automotive steering linkages, *ASME J. Mech. Design*, 119, 481-484. https://doi.org/10.1115/1.2826393.

[2] S. Pramanik. 2002. Kinematic synthesis of a six-member mechanism for automotive steering, *ASME J. Mech. Design*, 124, 642-645. https://doi.org/10.1115/1.1503372.

[3] A.D. Juan, R. Sancibrian and F. Viadero. 2012. Optimal synthesis of function generation in steering linkages, *Int. J. Automotive Tech.*, 13(7), 1033-1046. https://doi.org/10.1007/s12239-012-0106-4.

[4] J.S. Zhao, X. Liu, Z.J. Feng and J.S. Dai. 2013. Design of an Ackermann-type steering mechanism, *Proc. IMechE Part C: J Mech. Engg. Sci.*, 227(11), 2549-2562.

[5] M.M. Ettefagh and M.S. Javash. 2014. Optimal synthesis of four-bar steering mechanism using AIS and genetic algorithms, *J. Mech. Sci. and Tech.*, 28(6) 2351-2362. https://doi.org/10.1007/s12206-014-0526-3.