行政院國家科學委員會補助專題研究計畫成果報告

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※ **扭轉彈簧可變彈性機構的研發** ※

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※ Development of a Variable Stiffness Mechanism ※

※ for Torsion Spring ※

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計畫類別：X□**個別型計畫** □**整合型計畫**

計劃編號 : NSC 89-2212-E-009 –010 -

.**執行期間** : 88 **年 8 月 1 日至 89 年 7 月 31 日**

計劃主持人 : **徐文祥**

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□ **赴國外出差或研習心得報告一份**

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執行單位：國立交通大學機械系

中 華 民 國 89 **年** 7 **月** 31 **日**

行政院國家科學委員會專題研究計畫成果報告

扭轉彈簧可變彈性機構的研發

Development of a Variable Stiffness Mechanism for Torsion Spring

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計劃主持人 : **徐文祥　國立交通大學機械系教授**

計畫參與人員：吳青台 陳信志 國立交通大學機械系研究生

一、中文摘要

後變速器是一變速自行車變速系統中重要的零組件之一。由過去相關研究中，已知平面後變速器中 B、P 彈簧之彈性係數值大小對導輪定位及對鍊條張力大小有直接的影響。

在本研究中，所模擬出平面後變速器之 B、P 彈簧之彈性係數變化對導輪定位的影響，在給定平面後變速器 P 彈簧不同彈性係數值的情況下，作一實驗與理論的比較。並利用一裝置，在變速系統定轉速下，測得緊邊鏈條的張力變化情形。證實當 P 彈簧彈性係數較小時，能使導鏈導輪與較大齒盤的間距與在較大 P 彈簧彈性係數時的導鏈導輪與較小齒盤的間距相等。

故在此我們先提出三種彈性係數可變機構，其特點是在扭轉角度增大時，其彈性係數可調小。並從製造及分析上比較其優劣點，最後決定其中一個較佳的設計，並針對此機構，製作一原型件進行測試，與分析結果比較及討論。

英文摘要

The rear derailleur is one of the key components in the derailed system of the multi-speed bicycle. In the past researches, it was found that the values of the B-spring stiffness and P-spring stiffness affect the tension of the chain and the positions of the guide pulley directly.

According to the simulation and experimental investigations on the influence of the B-spring and P-spring stiffness to the rear derailleur system, it is found that the smaller P-spring stiffness at larger torsion angle can keep the distance between the guide pulley and the freewheel more close to a constant in shifting.

Therefore, three variable stiffness mechanisms are proposed first. Then the advantage and the disadvantage are compared to decide a better design. For such mechanism, the prototype is manufactured, analysized, and tested.

二、計劃緣由與目的

The function of the rear derailleur is to move the chain from one cog to another, and the rear derailleur should guide the chain rapidly, smoothly and precisely to a selected rear sprocket. So the rear derailleur is one of the key components in a multi-speed bicycle.

A few patents (Yamasaki, 1980; Nagano, 1980, 1986, 1988, 1989; Jona, 1987; Romano, 1989; Testa, 1989) concerning the improving rear derailleur systems have been issued. Those designs all want to make the path of the guide pulley parallel to the conic profile of the freewheel sprockets to satisfy the requirement of the distance between the guide pulley and each freewheel sprocket are constants by different mechanical design.

It will be easy to see the disadvantage of the planar rear derailleur. The guide pulley's path of planar rear derailleur doesn't satisfy the requirement keeping the same distance between the freewheel and the guide pulley in every speed-ratio. So it is easy to hit the bigger freewheel sprocket.

In order to design a rear derailleur with high capacity, the teeth number of every freewheel sprocket must vary a lot, and the dimension of every freewheel sprocket will vary concurrently in the limited space of the freewheel body. So the shape of freewheel sets may no longer form a cone again. Such shape of the freewheel allows the cyclist to down-shift faster. This shape brings another problem in designing the path of the rear derailleur to mesh such freewheel.

The guide pulley's positions are affected by the ratio of B-spring and P-spring from mathematical model in the planar rear derailleur (Lee 1997). Therefore, if the ratio of B-spring and P-spring can be varied at different speed-ratios, the position of the guide pulley may be tuned to generate different path.

Here a variable stiffness mechanism is designed, analyzed, and tested to provide a smaller spring rate at larger torsional angle for the rear derailleur which has not been reported before.

三、研究方法

A static mathematical model of the planar rear derailleur was built by Lee(1997). The results from simulation and experiment in each speed-ratio were also compared and discussed. The relations between two spring constant ratio and the positions of the guide pulley and tension pulley were also discussed by simulation.

The stiffness of the B-spring and P-spring will indeed vary the position of the guide pulley, also the value of the tight-side tension force. By the experimental results, the mathematical model is validated that the reliability of the simulated results, and the goal of the variable stiffness mechanism is clear.

The different spring stiffness can affect the positions of the guide pulley. Thus, an adjustable stiffness mechanism may change the path of the guide pulleyand keep the distance between the sprocket and guide pulley closer to a constant at different gear ratio. Then, the spring rate is required to be smaller at large torsion angle. Different schemes are proposed here including springs in parallel or serial connection.

In general, the variable spring stiffness can be divided into two phenomena as Figure 1 shown. The combination of two more springs in serial connection or in parallel connection can make these two phenomena realizing.

The spring stiffness can be affected by different spring connection. If two springs are in serial connection, the stiffness will become lower. If two springs are in parallel connection, the stiffness will become larger. Therefore, if two springs are not connected at initial working range, then switch to serial connection at larger working range, a variable stiffness at different working range can be achieved, as shown in Fig. 1.

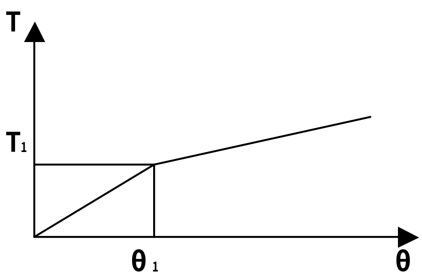

Figure 1. Two different stiffness at different working ranges

四、 結果與討論

As shown in Figure 2 and 3, the proposed mechanism is composed with following parts: base, cylinder, main spring, auxiliary spring, rotated disk, axis, and pre-load pin. The auxiliary spring needs to be turned a required angle first. The pre-torque can be calculated by the spring stiffness and relative angle (Hook's Law). The auxiliary spring and the main spring are in serial connection through a rotated disk. The auxiliary spring can be treated as a rigid-body when the applied torque in the main spring is less than the pre-torque. When the applied torque working in the main spring is larger than the pre-load in the auxiliary spring, both main spring and the auxiliary spring will function. Then two springs become in serial connection. The stiffness of this spring mechanism can be written as $K = K_1 * K_2 / (K_1 + K_2)$, as shown in Figure 1.

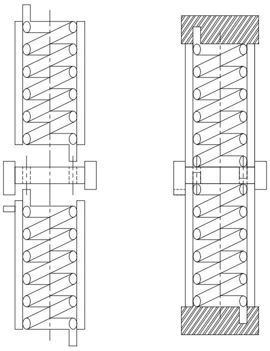

Figure 2 Concept design of the variable stiffness mechanism

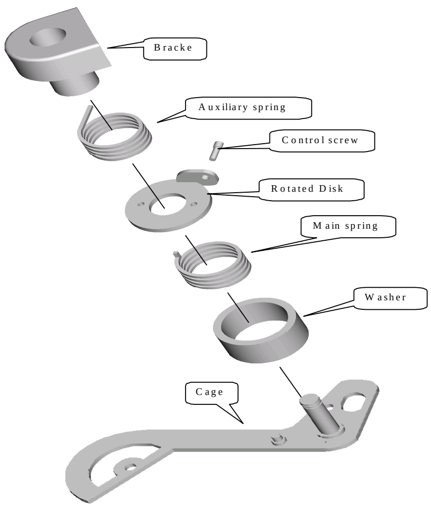

Figure 3. Components in the proposed variable stiffness mechanism

In order to verify the design concept, the proposed variable spring mechanism is fabricated, tested and simulated. The testing facility is set up by the Mechanical Transmission Components Department of the Mechanical Industry Research Laboratories in the Industrial Technology Research Institute. An overall view of the testing facility is shown in Figure 4.

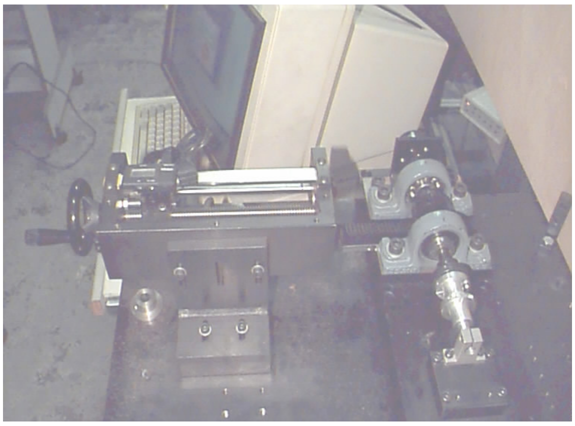

Figure 4 Testing facility

In testing and simulation, the spring with 1.2 mm wire diameter (spring I) is set the auxiliary spring. Other springs with 1.4 mm (spring II), 1.6 mm (spring III), and 1.8 mm (spring IV) wire diameter are set as the main spring separately in each testing.

The springs in testing may be affected by the outside constraints for example frictional torque in the mechanism. Because the friction affects the stiffness of the mechanism, the curve of the testing results can't match the curve of the simulated results. The relation between torque and rotated angle before connection can be expressed as

$$T_{b.c.} = K_M \times \Delta^{-}{}_{c} + T_{fm}$$

and the torque equation after connection can be expressed as

$$T_{a.c} = K_{serial} \times \Delta^{-}{}_{c} + T_{fd}$$

$T_{b.c.}$：the torque before two springs in serial connection
$T_{a.c.}$：the torque after two springs in serial connection
T_{fm}：the maximum static friction torque
T_{fd}：the dynamic friction torque
$\Delta\theta_c$：the rotated angle of two springs in serial connection

The comparison between testing and simulated results are made, and one of the results is shown in Fig. 5.

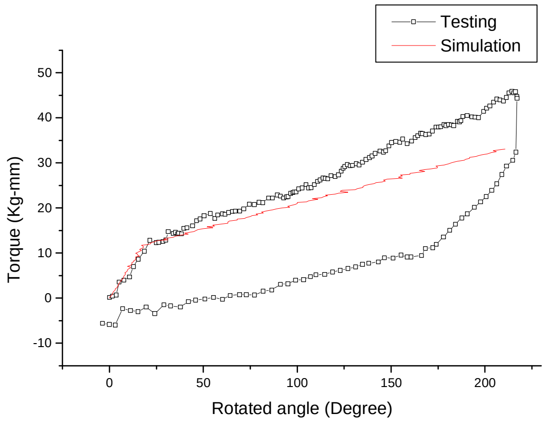

Figure 5 Spring I and IV in serial connection with 90° pre-load angle

The harder spring stiffness is applied in this testing to decrease the frictional effect. As Figure 22 shown, the difference of testing and simulated results are decreased, and the frictional torque is not constant in this spring mechanism. The frictional torque is increased as the rotated angle is increased.

By the comparisons of results, the curve of Torque-Rotated Angle is similar to the variable spring mechanism design. There is still a little difference between simulated and testing results. The differences are generated by outside constraints. When the spring is rotated, the spring bulking and the length variation of the spring. Because the phenomenon of the spring bulking, the spring will hit the field wall to release the loops of working spring. It will make the spring stiffness harder. The spring length will be longer as the torsion angle is larger. However, the spring space doesn’t change, it will make the normal force larger, and two body’s surface friction are also larger. The difference of the testing spring stiffness can be improved by mending outside constraints for example assembling bearing in each contact surface or release the hitting opportunity between spring and wall.

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