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Research Article

Analysis for Suspension Hardpoint of Formula SAE Car Based on Correlation Theory

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Abstract: According to the correlation of the hardpoints, double wishbone suspension of Formula SAE Car can be improved and optimized. For Formula SAE car, performance and kinetic characteristic of the suspension follow very closely to handling stability, comfort ability, handling of steering and service life of tire, etc. Double wishbone suspension is chosen as front suspension of Formula SAE car in the paper. Based on the multi-body dynamics and suspension kinematics theory, simulation model of front suspension is built and analyzed. With correlation theory, the correlation for the hardpoints of the suspension is studied and discussed. Through all the testing, not only a lot of four wheel alignment parameter variations can be obtained, but also the rational designing for hardpoints which can provide an useful reference for vehicle designing.

Keywords: Correlation theory, double wishbone suspension, formula SAE car, simulation model

INTRODUCTION

The suspension system plays an important role with regard to the performance of a vehicle in terms of its handling ability, stability, and ride comfort. Double wishbone independent suspension is widely used on automobile now. Two wishbones have equal length or not. Equal length of double wishbone independent suspension is not usually used now. Unequal length of double wishbone independent suspension can keep good road ability and reduce the interference between suspension and steer bar, with reasonable structural parameters and proper arrangements to make the parameter of wheel spin and wheel location floating in permissible range. Stabilization of Formula SAE Car with high speed is more important than ride comfort (William and Douglas, 1995). Considering space and placement of Formula SAE Cars, double wishbone suspension is chosen in the countries all over the world. Unequal length of double wishbone suspension can keep good stabilization and have suitable roll center and longitudinal tipping Center with the right hardpoints and the length of control arm (Shun-Kai *et al*., 2006).

Modeling and simulation of suspension systems exist in a significant number of publications, for example, M´antaras *et al*. (2004) and Cronin (1981). Similarly, the double wishbone suspension is modeled and simulated dynamically (Attia, 2001; Liu *et al*., 2010). There are very few papers to analyze and discuss the design and optimization of the suspension system. The existing suspension systems redesigned or optimized by eliminating or reducing a certain defect or fault inherent during the suspension design (Bian *et al*., 2004; Habibi *et al*., 2007). A better suspension system can be obtained by improving or optimizing the existing and working suspension. The effect of the suspension kinematics on the vehicle handling characteristics and perform vehicle handling simulations is studied (Lee *et al*., 2001; Makita, 1999). The kinematic model of the suspension is actively applied to obtain optimum ride and handling characteristics. Apparently, there have not been focused efforts on methods to design and optimize suspension for quantitative ride characteristics. The conclusion is some hardpoints will have heavy effect on suspension characteristic and performance by the hardpoint correlation. The hardpoint correlation can provide a reference for improvement or optimization of the suspension.

Front double wishbone independent suspension is regard as research object (Li *et al*., 2003; Liu *et al*., 2011), using multi-body dynamics and suspension kinematics theory to analyze and discuss the correlation for the hardpoints of the suspension. With the correlation of the hardpoints, the performance of front suspension can be improved and optimized.

DOUBLE WISHBONE FRONT SUSPENSION MODEL

There are 12 Hardpoints in the double wishbone front suspension of Formula SAE (Badih and Brian, 2002). One-half front suspension has 6 Hardpoints as follows in Fig. 1. Connection points between upper control arm and the frame are hardpoint 1 and 2.

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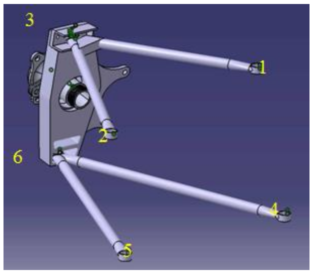

Fig. 1: One-half front suspension

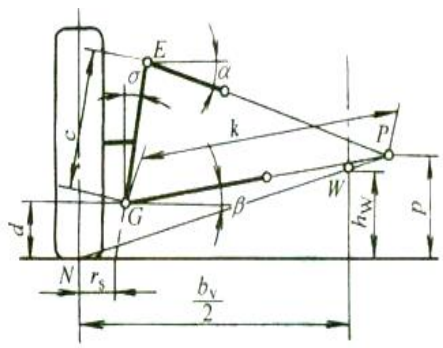

Fig. 2: Parameters of unequal length front suspension

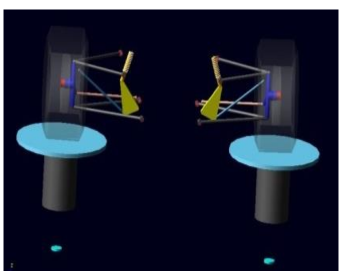

Fig. 3: Front suspension simulation model

Table 1: The hardpoints of front suspension

The hardpoint	X (mm)	Y (mm)	Z (mm)
Hardpoint 1	440.988	-288.526	205.629
Hardpoint 2	828.389	-288.526	201.415
Hardpoint 3	521.200	-581.044	232.087
Hardpoint 4	303.347	-171.947	18.200
Hardpoint 5	701.268	-171.947	18.200
Hardpoint 6	506.500	-581.044	22.010

Connection points between lower control arm and the frame are hardpoint 4 and 5. Connection point between upper control arm and the column is hardpoint 3. Connection point between lower control arm and the column is hardpoint 6. Hardpoint 3 and 6 are obtained with the length of kingpin and wheel alignment parameters. All other hardpoints can be used to decide the parameters of double wishbone suspension and its performance. The placement of six hardpoints affects suspension performance and stabilization.

The length of upper control arm or lower control arm has a great effect on wheel alignment parameters. Racing wheel tread should have small change when wheel travel, which can reduce the wear of the tire and extend its serviceable life. So that length ratio between upper control arm and lower control arm is about 0.6. To keep good stabilization, racing wheel alignment parameters should have small change. At this moment length ratio between upper control arm and lower control arm is about 1.0. To sum up the above arguments, the range of length ratio is 0.6 to 1.0.

Front suspension model of Formula SAE Car is built based on multi-body dynamics and Suspension Kinematics theory in Fig. 2. The hardpoints of front suspension is as follows in Table 1.

Front suspension simulation model are built and showed as in the Fig. 3. Changes of wheel alignment parameters are directly affected by design rationality of guiding mechanism in an independent suspension with double wishbone. To meet racing car's performance need, Changes of wheel alignment parameters should be kept within reasonable bounds. Wheel alignment parameters are analyzed and discussed in this paper, for example, camber angle, caster angle, kingpin angle and toe angle.

HARDPOINTS CHANGE ANALYSIS OF FRONT SUSPENSION

Hardpoints change related to X axis: The value of hardpoint 1 is changed at X-axis incrementally by δx 2mm, while keeping the other hardpoints unchanged. During the move of hardpoint 1, the change of hardpoint 1 maybe interferes with shock absorber and the rocker. The camber changes about 0.01°, caster with 0.006°, kingpin 0.01° and toe-in 0.04°. In a word, the move of hardpoint 1 has slight effect on wheel alignment parameters.

The value of hardpoint 2 is changed at X-axis incrementally by δx 20 mm, while keeping the other hardpoints unchanged. The change of hardpoint 2 has a wide range at X-axis. During the move of hardpoint 2, the camber changes about 0.02°, caster with 0.015°, kingpin 0.01° and toe-in 0.004°. where x=728.389, there are abnormal value of caster 0.1888, but all other parameters have slight change. So that x = 728.389 is not normally present in the hardpoint design. In a word, the move of hardpoint 2 has slight effect on wheel alignment parameters.

The value of hardpoint 4 is changed at X-axis incrementally by δx 2 mm, while keeping the other hardpoints unchanged. During the move of hardpoint 4, the change of hardpoint 4 maybe interferes with shock absorber and the rocker. During the move of hardpoint 4, the camber changes about 0.001°, caster with 0.002°, kingpin 0.01° and toe-in 0.001°. In a word, the move of hardpoint 4 has slight effect on wheel alignment parameters.

The value of hardpoint 5 is changed at X-axis incrementally by δx 2 mm, while keeping the other hardpoints unchanged. During the move of hardpoint 5, the change of hardpoint 5 maybe interferes with shock absorber and the rocker. During the move of hardpoint 5, the camber changes about 0.001°, caster with 0.001°, kingpin 0.001° and toe-in 0.002°. In a word, the move of hardpoint 5 has slight effect on wheel alignment parameters.

As stated above, four hardpoints are changed at X-axis. And the move of the hardpoints has slight effect on wheel alignment parameters. Then suspension hardpoints can be changed according to hardpoint layout because of slight performance change for wheel alignment parameters.

Hardpoints change related to Y axis: The value of hardpoint 1 is changed at Y-axis incrementally by δY 3mm, while keeping the other hardpoints unchanged. Since the Y-value would have impact on the length of rocker arm, while the length ratio of the two arms should be set between 0.6-0.8, the allowance for changes should be controlled between ±30mm. When the y-value increase, the length of front upper wishbone would increase and camber decrease with ∆max equals to 0.2253. In the meanwhile, the caster would incline to decrease with the maximum ∆value of 0.015, so does the kingpin with ∆max = 0.2075. However, the toe-in would increase instead with ∆max = 0.2376. Thus, the Y value for hard point 1 should fall into set [258.526, 288.526] considering of the comparatively reasonable solution for caster design and changing four-wheel alignments.

The value of hardpoint 2 is changed at Y-axis incrementally by δy 3 mm, while keeping the other hardpoints unchanged. Since the Y-value would have impact on the length of rocker arm, while the length ratio of the two arms should be set between 0.6-0.8, the allowance for changes should be controlled between ±30mm. When the y-value increase, the length of rear upper wishbone would increase and camber decrease with ∆max equals to 0.0514. In the meanwhile, the caster would incline to decrease with the maximum ∆value of 0.0069, so does the kingpin with ∆ max = 0.0479. However, the toe-in would increase instead with ∆max = 0.0645. Thus, the Y value for hard point 2 should fall into set [258.526, 300.526] considering of the comparatively reasonable solution for caster design and changing four-wheel alignments.

The value of hardpoint 4 is changed at Y-axis incrementally by δy 3mm, while keeping the other hardpoints unchanged. Since the Y-value would have impact on the length of rocker arm, while the length ratio of the two arms should be set between 0.6 -0.8, the allowance for changes should be controlled between ±30mm. When the y-value increase, the length of front lower wishbone would increase and camber increase with ∆max equals to 0.0118. In the meanwhile, the caster would incline to decrease with the maximum ∆value of 0.0479, so does the kingpin with ∆max = 0.0116. However, the toe-in would increase instead with ∆max = 0.006. Thus, the Y value for hard point 4 should fall into set [141.947, 201.947] considering of the comparatively reasonable solution for caster design and changing four-wheel alignments.

The value of hardpoint 5 is changed at Y-axis incrementally by δy 3 mm, while keeping the other hardpoints unchanged. Since the Y-value would have impact on the length of rocker arm, while the length ratio of the two arms should be set between 0.6 -0.8, the allowance for changes should be controlled between ±30mm. When the y-value increase, the length of rear lower wishbone would increase and camber decrease with ∆max equals to 0.0013. However, the caster would incline to increase with the maximum ∆value of 0.0514, so does the kingpin with ∆max = 0.0061. In the meanwhile, the toe-in would increase instead with ∆max = 0.057. Thus, the Y value for hard point 5 should fall into set [141.947, 201.947] considering of the comparatively reasonable solution for caster design and changing four-wheel alignments.

As stated above, four hardpoints are changed at X-axis. Hard piont 1 has heavy effect on wheel alignment parameters. But the move of all other hardpoints has slight effect on wheel alignment parameters.

Hardpoints change related to Z axis: The value of hardpoint 1 is changed at Z-axis incrementally by δz 3mm, while keeping the other hardpoints unchanged. As the allowance for changing of Z value is larger than on other directions, the allowance can be set between ±30 mm. With the Z value increases, the front upper wishbone would slant and the camber would decline with the maximum change at 2.0067, caster increase with ∆max = 0.1995, the kingpin decrease with ∆max = 2.1127, and toe-in increase by maximum 1.9495. When hardpoint 1 moves between [175.629，205.629] at Z-direction, the camber would not change in according with the compression and explanation of springs. In this circumstance, danger of unnecessary damage of kingpin would occur during driving of the vehicle if this design would be utilized. Thus, the most suitable interval for Z-value should be set between [205.629, 220.629] for hardpoint 1.

The value of hardpoint 2 is changed at Z-axis incrementally by δz 3mm, while keeping the other hardpoints unchanged. As the allowance for changing of Z value is larger than on other directions, the allowance can be set between ±30 mm. With the Z value increases, the rear upper wishbone would slant and the camber would decline with the maximum change at 0.6116, caster increase and then decrease, the kingpin decrease with ∆max = 0.5431, and toe-in increase by maximum 1.002. When hardpoint 2 moves between [201.415，231.415] at Z-direction, the camber would not change in according with the compression and explanation of springs. In this circumstance, danger of unnecessary damage of kingpin would occur during driving of the vehicle if this design would be utilized. Thus, the most suitable interval for Z-value should be set between [180.415, 201.415] for hardpoint 2.

The value of hardpoint 4 is changed at Z-axis incrementally by δz 3mm, while keeping the other hardpoints unchanged. As the allowance for changing of Z value is larger than on other directions, the allowance can be set between ±30 mm. With the Z value increases, the front lower wishbone would slant and the camber would increase with the maximum change at 0.9992, caster increase and then decrease, the kingpin decrease with ∆max = 1.3379, and toe-in decrease and then increase. When hardpoint 4 moves between [18.2， 48.2] at Z-direction, the camber would not change in according with the compression and explanation of springs. In this circumstance, danger of unnecessary damage of kingpin would occur during driving of the vehicle if this design would be utilized. Thus, the most suitable interval for Z-value should be set between [0.2, 18.2] for hardpoint 4.

The value of hardpoint 5 is changed at Z-axis incrementally by δz 3mm, while keeping the other hardpoints unchanged. As the allowance for changing of Z value is larger than on other directions, the allowance can be set between ±30mm. With the Z value increases, the rear lower wishbone would slant and the camber would increase with the maximum change at 0.4852, caster increase and then decrease, the kingpin increase with ∆max = 0.7498, and toe-in decrease and then increase. When hardpoint 5 moves between [-12.2, 18.2] at Z-direction, the camber would not change in according with the compression and explanation of springs. In this circumstance, danger of unnecessary damage of kingpin would occur during driving of the vehicle if this design would be utilized. Thus, the most suitable interval for Z-value should be set between [18.2, 24.2] for hardpoint 5.

Stated thus, 4 hardpoints are changed at X-axis. All the hardpoints has heavy effect on wheel alignment parameters.

1-2 and 4-5 change related to Y axis: Change the Y-value of hardpoint 1 and hardpoint 2 incrementally by ∆x = 3 mm every time with other points unchanged, the equalized amount of change implies that the length of the upper control arm would change accordingly, which would have a bigger impact on the four-wheel alignment parameters. Therefore，a validation test of comparison between the upper and lower control arms should be conducted to obtain the rational parameters for the upper control arm.

Change the Y-value of hardpoint 4 and hardpoint 5 incrementally by ∆x = 3 mm every time with other points unchanged, the equalized amount of change implies that the length of the lower control arm would change accordingly, which would have a bigger impact on the four-wheel alignment parameters. Therefore，a validation test of comparison between the upper and lower control arms should be conducted to obtain the rational parameters for the lower control arm.

1-2 and 4-5 change related to Z axis: Change the value of hardpoint 1 and hardpoint 2 at Z direction with a 3 mm margin every time with other points unchanged, the slant of upper cross arm would affect four-wheel alignment positioning performance. However, when a small caster change occurs, the proper position can be detected when the Z value of the front and rear point is close enough based on the previous test.

Change the value of hardpoint 4 and hardpoint 5 at Z direction with a 3mm margin every time with other points unchanged. According to the analyses, the conclusion is consistent with the change of hardpoint 1 and hardpoint 2 at Z direction.

The change for hardpoint 3 and 6: Hard point 3 and 6 have heavy effect on the column location and wheel tread. Because hardpoint 3 and 6 have heavy effect on the layout of racing car at X-axis and Z-axis, hardpoint 3 and 6 are not analyzed and discussed. Change the value of hardpoint 3 and hardpoint 6 at X direction with a 2mm margin every time with other points unchanged. When wheel tread is increased, camber has small change. However caster has a slight change, kingpin decrease, and toe increase. In a word, considering suspension performance and racing trafficability, unchanged front tread can be suitable.

THE ANALYSIS AND DISCUSSION OF THE HARDPOINTS CORRELATION

As stated above, with correlation theory, the correlation for the hardpoints of the suspension is studied and discussed. The result for the correlation of the hardpoints is shown as in the Table 2.

According to Table 2, the correlations between the changes of hardpoint 1, 2 are high for steering axis inclination, camber and toe-in at Y direction, which means that no matter how the hardpoints change would not affect the four-wheel alignment parameters. Since the change of caster would affect the four-wheel alignment parameters, the change of caster would be minimized when the Y value of hardpoint 1 and 2 are close enough to get the proper caster.

For hardpoint 4 and 5, the changes of value at Y-axis would have a bigger impact on the four-wheel alignment parameters based on the correlation perspective. However, the absolute amount of the parameters does not change a lot besides the camber, which shows a better results when the Y-value of hardpoint 5 is less than that of hardpoint 5.

When moves the upper cross arm (hardpoint 1-2) and lower cross arm (hardpoint 4-5), it is shown similar results that the co-efficiencies are high enough to prove the significances except the caster. Since the changes for Y-values of the upper and lower cross arms would be equal to the changing of upper and lower cross arms ratio, it should be suitable to fit into the interval of 0.6-0.8.

Table 2: The correlation of the hardpoints

	Camber	Caster	Kingpin	Toe
Hardpoint 1and 2 at Y-axis	0.995434	0.692997	0.995665	0.996698
Hardpoint 1 and 1-2 at Y-axis	0.99743	0.002895	0.999696	0.999804
Hardpoint 2 and 1-2 at Y-axis	0.991326	-0.13107	0.99308	0.994909
Hardpoint 4 and 5 at Y-axis	0.916882	-0.99987	0.996255	0.885703
Hardpoint 4 and 4-5 at Y-axis	0.986685	-0.99597	0.987886	0.832776
Hardpoint 5 and 4-5 at Y-axis	0.867965	-0.99606	0.972537	0.994346
Hardpoint 1-2 and 4-5 at Y-axis	-0.99541	0.049924	-0.997800	0.999909
Hardpoint 1 and 2 at Z-axis	0.988715	0.850493	0.999991	0.998651
Hardpoint 1 and 1-2 at Z-axis	0.992243	0.429646	0.999869	0.96355
Hardpoint 2 and 1-2 at Z-axis	0.990393	-0.0439	0.999883	0.952362
Hardpoint 4 and 5 at Z-axis	0.99997	0.894976	0.999972	0.98591
Hardpoint 4 and 4-5 at Z-axis	0.999843	-0.36577	0.999989	0.822081
Hardpoint 5 and 4-5 at Z-axis	0.999856	0.036961	0.999983	0.897772
Hardpoint 4 and 4-5 at Z-axis	-0.99697	0.939289	-0.99974	0.817461

When referring to the movements on Z-axis for hardpoint 1 and 2, the changes are significant enough except the caster, which is similar with other results. Based on the data listed at the Table 2, when the Z-value decreases at hardpoint 1, a better performance is shown which would coincide with the theories for hardpoints arrangements. Meanwhile, based on the correlations of these parameters, it is much more stable for caster when decreases the Z-values of the two points in the same time.

The findings also show the same results when changing the values of hardpoint 4 and 5 at Z-direction. When reached out the maximum for distance changes, some of the four-wheel alignment parameters would change 2° at most. Based on data shown at Table 2, it would show a better result when decreasing the Z-value of hardpoint 4. Like hardpoint 1 and 2, decreasing the values of point 4 and 5 simultaneously would generate a more satisfied result.

Based on Table 2, it is showed that the parameters displayed when the hardpoint 1-2 moves up and point 4-5 down at Z-axis would fit the requirements for designing a Formula SAE Car. At this circumstance, the upper and lower cross arms are almost parallel with each other which means that when the 3 hardpoints of the upper arm and 3 points at the lower cross arm can exert a flat surface or parallel each other would produce the beast performance for suspension system of the vehicle.

CONCLUSION

It is addressed that how a Formula SAE car front suspension is designed based on calculations of hardpoints of the suspension based on kinematic simulation. The data generated by the software would provide evidences and directions for the production process.

Firstly, a double wishbone independent suspension as the front suspension system chosen based on the requirements and characteristics of Formula SAE cars. Secondly, optimization of front suspension hard points arrangements: the virtual simulation model for the front suspension system of Formula SAE is built. Then, the hardpoints arrangements are changed each time and the data of parameter variations is obtained at different conditions with kinematics simulation analysis. Finally, the optimal arrangements for virtual can be obtained according to the correlation test of each hardpoint.

According to the hardpoint correlation, the parameters displayed when the hardpoint 1-2 moves up and point 4-5 down at Z-axis would fit the requirements for designing a Formula SAE Car. So the performance of the suspension can be improved and optimized by adjust the hardpiont 1, 2, 4 and 5 at Z-axis. A reference can be provided for improvement or optimization of the suspension with the hardpoint correlation. And the scheme of the suspension can be made based on the correlation and virtual experiment.

Through all the testing, not only a lot of four wheel alignment parameter variations can be obtained, but also the rational designing for hardpoints which can provide an useful reference for vehicle designing. For future events like SAE, players can continue to test virtual data by simulation model or conduct analysis both on virtual model and real car which can compare the tests to improve the designing process and validity.

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REFERENCES

Attia, H.A., 2001. Dynamic modelling of the double wishbone motor-vehicle suspension system. Europ. J. Mech., 21(1): 167-174.

Badih, A.J. and D.P. Brian, 2002. Design of formula SAE suspension components. Proceeding of the SAE Motorsports Engineering Conference and Exhibition, pp: 382.

Bian, X.L., B.A. Song and W. Becker, 2004. The optimisation design of the McPherson strut and steering mechanism for automobiles. Forschungim Ingenieurwesen, 68(1): 60-65.

Cronin, D.L., 1981. MacPherson strut kinematics. Mech. Mach. Theory, 16(6): 631-644.

Habibi, H., K.H. Shirazi and M. Shishesaz, 2007. Roll steer minimization of McPherson-strut suspension system using genetic algorithm method. Mech. Mach. Theory, 43(1): 57-67.

Lee, S.H., U.K. Lee and C.S. Han, 2001. Enhancement of vehicle handling characteristics by suspension kinematic control. Proc. Inst. Mech. Eng., 215: 197-216.

Li, X., Y. Zhang, L. Zhou and M. Peng, 2003. Characteristic analysis and calculation of wishbone independent suspension with torsion bar. Automot. Eng., 25(1): 15-19.

Liu, X., H. Huang, L. Zhao, W. Jichang, G. Hui, *et al*., 2010. Influence of front double wishbone independent suspension performance on rubber bushing stiffness of the wishbone. Mach. Design Manuf., 9: 116-117.

Liu, X., L. Zhao, H. Huang, Y. Wang and H. Chen, 2011. Influence of outer rubber bushing stiffness of the under control arm on multi-link independent suspension performance. Adv. Mech. Eng., 52-54: 578-582.

M´antaras, D.A., P. Luque and C. Vera, 2004. Development and validation of a three-dimensional kinematic model for the McPherson steering and suspension mechanisms. Mech. Mach. Theory, 39(6): 603-619.

Makita, M., 1999. An application of suspension kinematics for intermediate level vehicle handling simulation. Soc. Autom. Eng. Japan, 20(4): 471-477.

Shun-Kai, Y., S. Chuan-Xue, W. Shi-Ze and P. Ji, 2006. Analysis of kinematics and elastokinematics characteristics between multi-link and double wishbone suspension. Automobile Technol., 12: 5-8.

William F.M. and L.M. Douglas, 1995. Race Car Vehicle Dynamics. Society of Automotive Engineers, Warrendale, Pa, pp: 890, ISBN: 1560915269.