A Study on the Minimization of Braking Pull in the Early Design Stage of Leaf Spring Suspension by the Simulations of Leaf Spring 2D and Modelica Model

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Abstract

Although the rate of use of air suspension is increasing in terms of ride comfort, commercial trucks are still selling vehicles equipped with rigid axles and leaf spring suspension mainly due to high loading and rough road driving. Due to these structural characteristics, geometry errors inevitably occur according to vehicle driving loads, and it takes a lot of time to solve the braking pull problem due to geometry errors in the vehicle development stage. Therefore, this study studied a methodology that could predict geometry errors during vertical and braking loads of suspension steering system more reliably in the early stages of design and applied the methodology to develop vehicles to improve braking pull.

FEM 3D leaf spring simulation has the advantage of being able to check deformation and stress for various loads, but it has the disadvantage of taking a lot of modeling and simulation time to respond to repetitive design changes. On the other hand, FEM 2D leaf spring analysis can quickly and reliably predict leaf spring behavior under vertical load and braking load required in the early design stage review, and it can be used together with design review for stiffness, strength, and durability performance, so it has an advantage when used in conjunction with performance review with the steering system in the early design stage.

Therefore, this study compared and verified with 3D FEM analysis and system test measurement to determine a reliable 2D FEM modeling level and predicted the leaf spring behavior according to load conditions. FEM leaf modeling was performed from the 2D shape of each individual plate in the free camber state to the U-bolt tightening condition, so that the free camber state shape of the leaf spring assembly could be predicted, so that the behavior of the leaf spring assembly under vertical and braking loads could be predicted well even at the design stage before the leaf spring manufacturing. Based on this 2D model simulation, the geometry errors for various knuckle arm hp positions, draglink and pitman arm combinations were quickly and reliably reviewed. In addition, a kinematic model was generated using the basic library of open modelica to perform a kinematic review of the steering device, which was utilized to design

a steering suspension system that minimizes braking pull in the early stage of vehicle design.

1. INTRODUCTION

It is known that the vehicle pull is caused by the asymmetric characteristics of the vehicle and tire, the left and right deviations of the braking torque, the road gradient and etc. but in this paper, a methodology for designing a concept that minimizes the geometry error of the steering system that causes the braking pull of a commercial vehicle with leaf spring suspension is presented.

The structure of the steering system used in conjunction with the leaf spring suspension system of a commercial truck is generally composed of a ball nut type steering gear and a pitman arm drag link structure as shown in Figure 1, so that a geometry error which is the difference between knuckle arm HP trajectory and draglink rear HP trajectory is inevitably present in the structure. Therefore, it should be designed to minimize a geometry error with the steering system in the suspension motion range during vehicle driving.

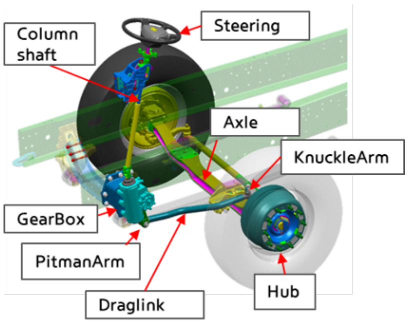


Figure 1: Leaf spring suspension and steering system.

However, design templates and ADAMS, the traditional methods of examining the geometric errors in the early stage of design, were not sufficient to examine them for each leaf spring assembly.

Since the design template examines the leaf spring behavior under vertical loads by modeling the leaf spring with a three-bar link, it may differ slightly from the actual leaf spring behavior depending on the shape of the leaf spring, and there are limitations in that the behavior under braking load cannot be examined.

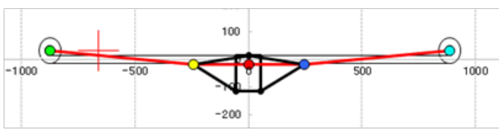


Figure 2: Leaf Spring Geometry Error Design Template

The ADAMS review was modeled based on the measurement data or the characteristics of the design goal rather than predicting the stiffness and strength performance of the leaf spring in the early stages of the design.

There is no significant error when reviewing the behavior of the leaf spring under vertical load, but since the detailed shape of the leaf spring is not reflected in all, it was not suitable to examine the geometric error due to the influence of the leaf spring windup behavior when applying braking load.

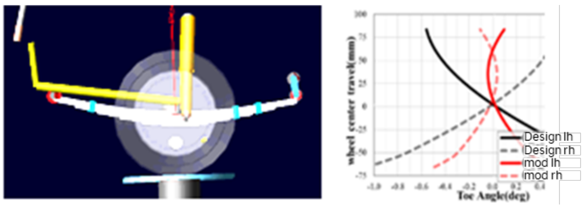


Figure 3: ADAMS Kinematic Review

The 3D FEM leaf spring simulation can predict the leaf spring behavior relatively accurately for various load conditions, but it takes a lot of modeling and simulation time to review various leaf spring designs in the early stages of design.

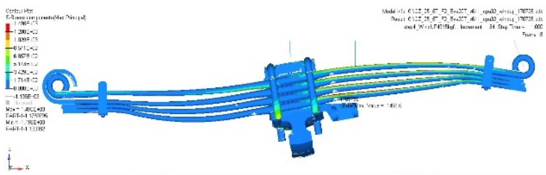


Figure 4: 3D FEM leafspring wind-up simulation

On the other hand, in the initial design stage review, 2D FEM leaf spring simulation can predict leaf spring behavior under vertical and braking loads rather than 3D FEM leaf spring simulation quickly and reliably. It also has the advantage of reviewing stiffness and strength of leaf spring in the initial design stage.

2. LEAF SPRING FEM 2D MODELING

Leaf spring 2D FEM simulation was performed using ABAQUS v2016 or higher version, which is a commercial solver and the 2D leaf spring modeling consists of individual plates, simplified U-bolts, shackles, bumper stoppers, Knuckle Arm HP, Wheel CTR HP, Tie rod HP, and Tire Patch HP. Each leaf is modeled on average as a 3 mm 2D element, with gaps in its initial state. The dead zone (U-bolt tightening area) of each leaf is held by kinematic coupling and connected between the individual leafs with a simplified U-bolt (1D element) as shown in Figure 5 below.

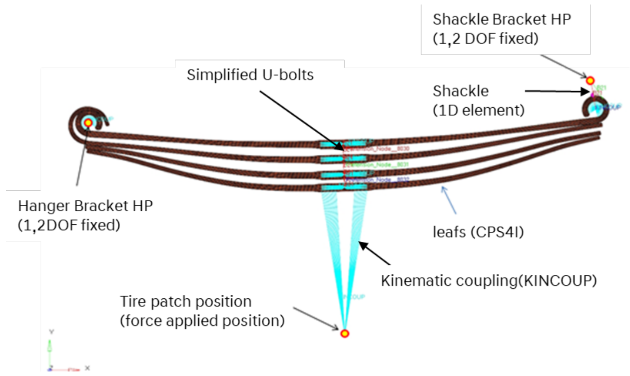


Figure 5: 2D leaf spring modeling

It varies depending on the 3D model, but when using a value of approximately 95% to 98% of the 3D leaf spring width, It was confirmed that stiffness and strength levels were implemented similarly to the 3D detailed model.

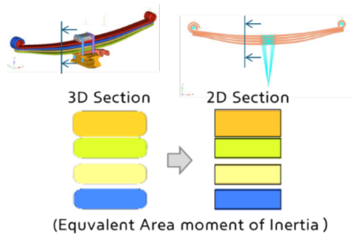


Figure 6: Section area of 2D leaf spring

the definition of contact between each leaf is set to general contact, and the contact between the shackle (1D beam element) and the first leaf is excluded using the *CONTACT EXCLUSIONS key word. Only degrees of freedom 1 and 2 are fixed to the hanger bracket mounting position and the shackle bracket mounting position, so that the degree of freedom of rotation is released. The simulation steps consist of U-bolt clamping step, a U-bolt fixed step, and an external load applying step. (See Fig. 7)

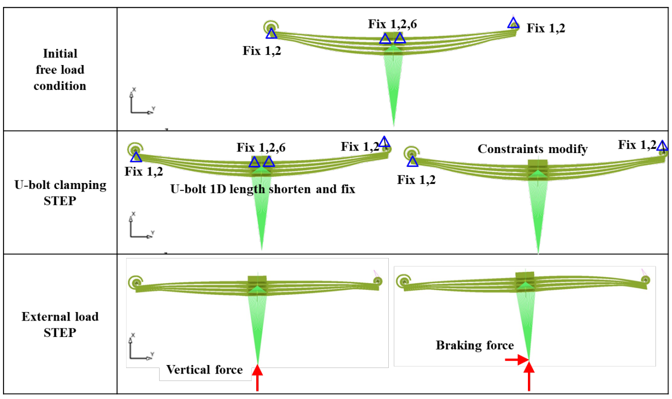


Figure 7: Steps in 2D FEM simulation

3. VALIDATION OF 2D FEM LEAF SPRING SIMULATION

A vertical 2G load application test was performed on the front axle leaf spring of a large truck, and as shown in figure 8,9 below, it was confirmed that the 2D element leaf spring simulation results matched the leaf spring strain measurement value relatively well.

Figure 8: Strain measurement TEST @ Vertical 2 GVW load

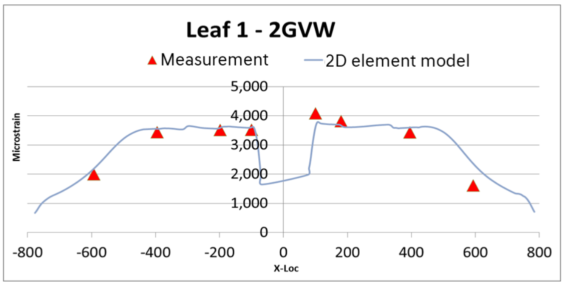


Figure 9: Comparison of TEST and 2D FEM Simulation (the first leaf)

and it was confirmed that the results of the 2D FEM simulation modeled based on the drawing information similarly reproduced the camber and stiffness level of the drawing. (See Figure 10, Table 1)

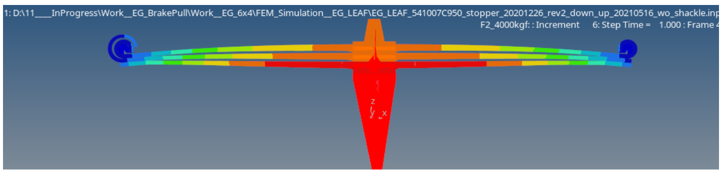


Figure 10: 2D FEM leaf spring simulation

	Drawing	Sim	Error (%)	Remarks
Free Camber	1	0.9992	0.08	Ratio
Kerb Camber	1	0.9856	1.44	Ratio
Leaf Stiffness	1	0.9953	0.47	Ratio (2000~4000kgf)

Table 1: Comparison of Drawing and 2D FEM Simulation

In addition, the leaf spring suspension test was performed under braking load conditions to confirm that the windup angle of the 2D FEM simulation was similar to the measured windup angle, and error of less than 2% were verified. (See Fig. 11, table 2)

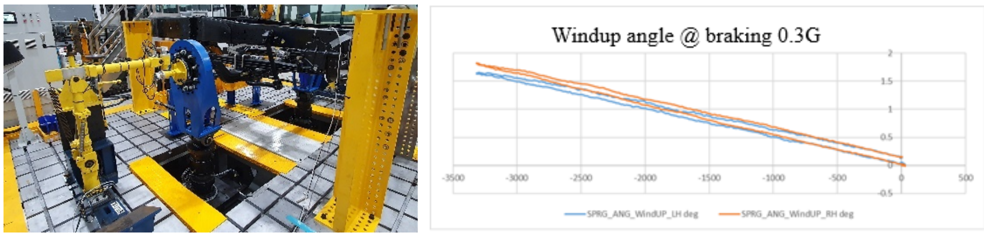
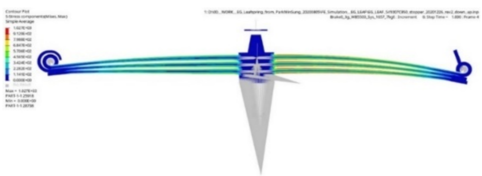


Figure 11: Wind-up angle measurement @ braking 0.3G



Simulation (degree)	LH,RH	1.77
TEST (degree)	LH	1.66
	RH	1.83
	average	1.74
Error(%)		**1.9**

Table 2: Comparison of TEST and 2D FEM simulation @ braking 0.3G

4. THE OPTICAL KNUCKLE ARM HP

Depending on the shape of the knuckle arm, there may be a difference between the trajectory of the knuckle arm hp in the vertical load range and the position of the knuckle arm hp in the braking load as shown in Figure 12 below. This difference can result in additional braking pulls in braking situations, even if the bump steer is small enough. Therefore, first of all, it is necessary to determine the optimal knuckle arm HP so that the location of the knuckle arm HP during braking can be as close as possible to the trajectory of the knuckle arm HP during vertical load.

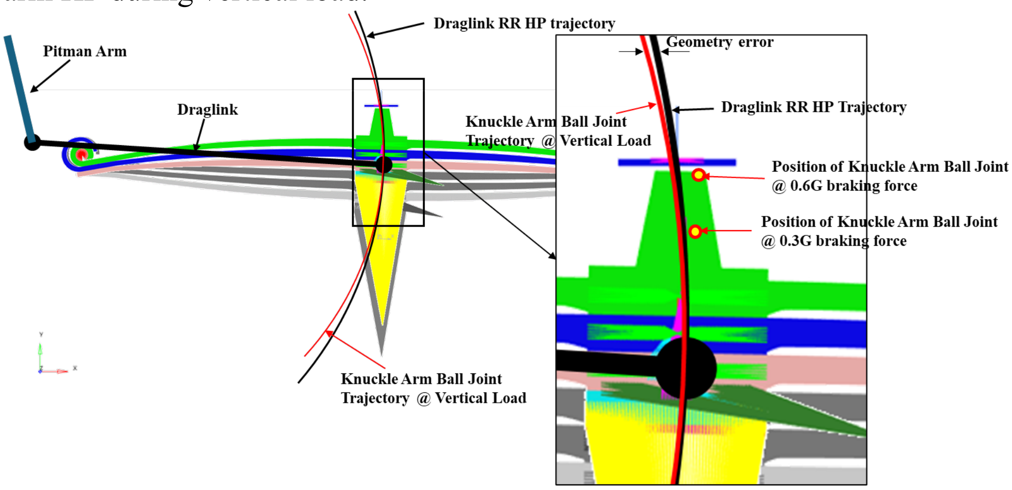


Figure 12: Geometry error between Knuckle Arm HP and Draglink RR HP Trajectory.

all available knuckle arm HPs were modeled as shown in Figure 13 below, and Through 2D FEM leaf spring simulation of vertical and braking load conditions, trajectory of each knuckle arm HP was reviewed as shown in Figure 14 below.

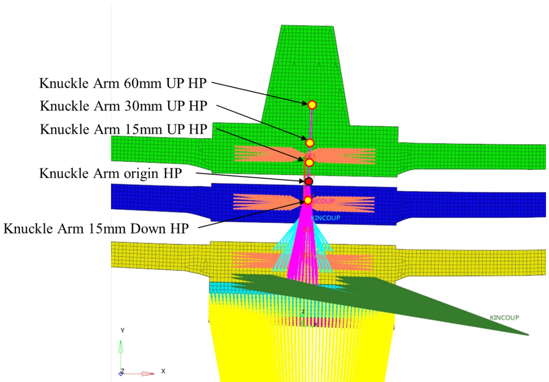


Figure 13: Knuckle Arm HPs modelling

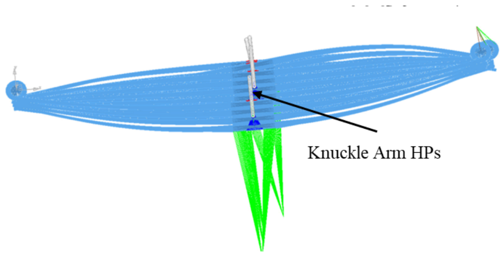


Figure 14: The trajectory of each knuckle arm H.P. according to load conditions

Once the optimal knuckle arm HP position has been determined, the next step can be to determine a steering system design that can minimize geometric error in the trajectory of the knuckle arm HP due to vertical and braking load leaf spring behavior and it can be reviewed using a kinematic model.

5. DEVELOPMENT OF KINEMATIC MODEL

In order to conduct a kinematic review with the suspension and steering system In order to conduct a kinematic review with the suspension and steering system based on the prediction results of the leaf spring behavior for each load condition using the 2D FEM model, a kinematic model that can be linked was needed, and the model was developed using the Mechanics library of OpenModelica as shown in Figure 15 below. Parameters were configured so that major hard points could be automatically changed according to changes in major design factors such as pitman arm length, drag link length, and pitman arm rotation axis position, and the kinematic model was modeled so that simulation results could be quickly checked for various parameter combinations.

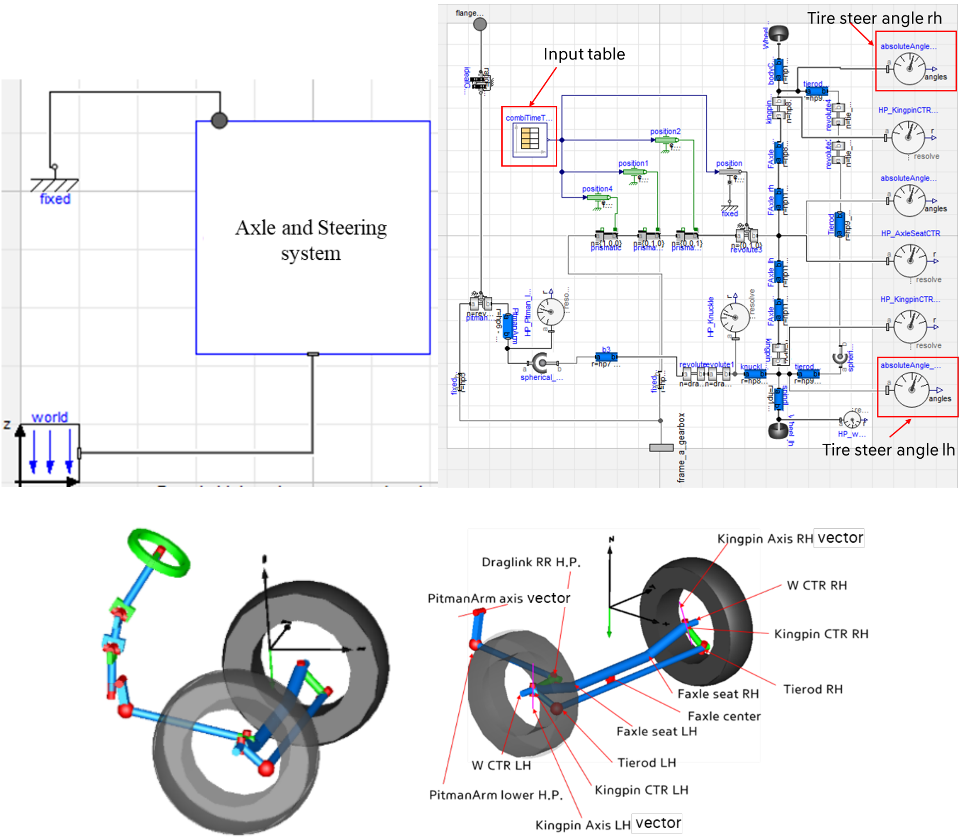


Figure 15: Open Modelica Modeling

The model is configured to check the tire steer angle by entering the front axle seat center position and axle angle information for each vertical load and braking load condition so that the 2D FEM simulation results can be linked to the open modelica model to quickly conduct a kinematic review.

Among the 2D FEM simulation results, front axle seat center position and axle angle information under each load condition used as input to the kinematic model were automatically organized in excel file using tcl code and python, and an Modelica Text was generated based on the excel file for a kinematic review. (See Fig. 16, table 4)

		FEM simulation results		
Step1	**clamp1**	**Axleseat ctr position(mm)**		**Axle angle(°)**
Step2	**clamp2**	**x**	**z**	
Step3	**Re_setting(FREE)**	-11.2	-667.3	-2.28
Step4	**KERB_load**	-3.6	-617.6	-2.16
Step5	**GVW_load**	-0.8	-581.9	-2.09
Step6	**Fz_4352.9kgf**	-0.3	-562.6	-2.08
Step7	**Fz_4815.7kgf**	-0.2	-552.6	-2.08
Step8	**Fz_6069.1kgf**	-1.0	-526.7	-2.11
Step9	**0.3g Braking**	4.1	-552.9	-0.41
Step10	**0.6g Braking**	9.6	-530.3	1.88

Table 3: 2D FEM simulation results for input of kinematic model

```
model EG6x4_Origin_Knuckle "EG6x4_Origin_Knuckle model"

parameter Real Rc=  "length of pitmanArm";
parameter Real Rs=  "length of draglink";
parameter Real rev_pitman[3] = { }   "rotate axis of pitmanArm";
parameter Real d = rev_pitman*(hp5-hp7);
parameter Real Cp[3] = hp7 + d*rev_pitman "center position of projection circle";
parameter Real Rp=sqrt(Rs*Rs-d*d);
parameter Real d2=Modelica.Math.Vectors.length(Cp-hp5) "distance between center of circle and center of projection circle of sphere";
parameter Real h = (1/2) + (((Rc*Rc) - (Rp*Rp))/(2 * (d2*d2)));
parameter Real Ri = sqrt(Rc*Rc - (h*d2)*(h*d2));
parameter Real Ci[3] = hp5+ h * (Cp - hp5);
parameter Real t[3]=Modelica.Math.Vectors.normalize(cross(Cp-hp5,rev_pitman));

parameter Modelica.SIunits.Position hp5[3] = { , , }   "hard point of pitmanArm upper";
parameter Modelica.SIunits.Position hp6[3] = Ci - t * Ri "hard point of pitmanArm lower";
parameter Modelica.SIunits.Position hp7[3] = { , , }   "hard point of Draglink RR";
parameter Modelica.SIunits.Position hp8_upr_lh[3] = { , , }   "hard point of Kingpin UPR LH";
parameter Modelica.SIunits.Position hp8_upr_rh[3] = {hp8_upr_lh[1],-hp8_upr_lh[2],hp8_upr_lh[3]}    "hard point of Kingpin UPR RH";
parameter Modelica.SIunits.Position hp8_lwr_lh[3] = { , , }   "hard point of Kingpin LWR LH";
parameter Modelica.SIunits.Position hp8_lwr_rh[3] = {hp8_lwr_lh[1],-hp8_lwr_lh[2],hp8_lwr_lh[3]}     "hard point of Kingpin LWR RH";
parameter Modelica.SIunits.Position hp8_1[3] = { , , }   "hard point of Kingpin CTR LH";
parameter Modelica.SIunits.Position hp8_2[3] = {hp8_1[1],-hp8_1[2],hp8_1[3]}     "hard point of Kingpin CTR RH";
parameter Modelica.SIunits.Position hp9_1[3] = { , , }   "hard point of Tierod LH";
parameter Modelica.SIunits.Position hp9_2[3] = {hp9_1[1],-hp9_1[2],hp9_1[3]}    "hard point of Tierod RH";
parameter Modelica.SIunits.Position hp10_1[3] = { , , }   "hard point of Wheel CTR LH";
parameter Modelica.SIunits.Position hp10_2[3] = {hp10_1[1],-hp10_1[2],hp10_1[3]}     "hard point of Wheel CTR RH";
parameter Modelica.SIunits.Position hp11_1[3] = {hp11_F1[1], ,hp11_F1[3]}  "hard point of Axle_seat_LH";
parameter Modelica.SIunits.Position hp11_2[3] = {hp11_F1[1], ,hp11_F1[3]}  "hard point of Axle_seat_RH";

parameter Modelica.SIunits.Position hp11_F0[3]= {-0.01116,0,-0.667294}   "hard point of AxleSeat center FREE";
parameter Modelica.SIunits.Position hp11_F1[3]= {-0.003596,0,-0.617614}   "hard point of AxleSeat center KERB";
parameter Modelica.SIunits.Position hp11_F2[3]= {-0.000812,0,-0.581859}   "hard point of AxleSeat center GVW";
parameter Modelica.SIunits.Position hp11_F3[3]= {-0.000263,0,-0.562549}   "hard point of AxleSeat center Fz_1";
parameter Modelica.SIunits.Position hp11_F4[3]= {-0.000241,0,-0.552618}   "hard point of AxleSeat center Fz_2";
parameter Modelica.SIunits.Position hp11_F5[3]= {-0.001043,0,-0.52674}   "hard point of AxleSeat center Fz_3";
parameter Modelica.SIunits.Position hp11_F6[3]= {0.004064,0,-0.552851}   "hard point of AxleSeat center Brake_0_3G";
parameter Modelica.SIunits.Position hp11_F7[3]= {0.009557,0,-0.530257}   "hard point of AxleSeat center Brake_0_6G";

parameter Real Axle_rad0= 0.039869   "F Axleseat rad FREE";
parameter Real Axle_rad1= 0.037622   "F Axleseat rad KERB";
parameter Real Axle_rad2= 0.036521   "F Axleseat rad GVW";
parameter Real Axle_rad3= 0.036334   "F Axleseat rad Fz_1";
parameter Real Axle_rad4= 0.036374   "F Axleseat rad Fz_2";
parameter Real Axle_rad5= 0.036794   "F Axleseat rad Fz_3";
parameter Real Axle_rad6= 0.007085   "F Axleseat rad Brake_0_3G";
parameter Real Axle_rad7= -0.032793   "F Axleseat rad Brake_0_6G";
          ⋮
end EG6x4_Origin_Knuckle;
```

Figure 16: Modelica Text of Kinematic model

By using this kinematic model, the amount of tire steer angle during braking is calculated for each design concept of the steering suspension system from the initial design stage of the vehicle, thereby selecting a concept design that reduces braking pull due to a geometry error compared to the existing mass-produced vehicle.

6. IMPROVENMENT CASE OF GEOMETRY ERROR OF LARGE TRUCK

Using previously developed FEM 2D simulation techniques and kinematic model, we conducted a kinematic review to derive a conceptual design in which 6x4 large trucks (spring span 1760mm) and 4x2 large trucks (spring span 1500mm) vehicles avoid braking pull due to geometric errors while using as many parts in common as possible, including knuckle arm, drag link, and pitman arm.

First of all, the smaller the difference between the design target load of the vehicle at the beginning of the design and the final vehicle weight, the higher the accuracy of the vertical and braking loads predicted at the design stage, so it was necessary to manage the vehicle's total weight and information such as front axles, knuckle arms, brake assemblies, tie rods, wheels & tires at each design stage.

The vertical load of the leaf spring was calculated by considering the unsprung mass for each design stage, and the braking load was calculated on the front leaf spring through the force equilibrium relationship as shown in Figure 17 below. In addition, the vehicle specifications required for the above force equilibrium were vehicle C.O.G height, wheelbase, front & rear axle weight, spring stiffness, and dynamic radius of tire.

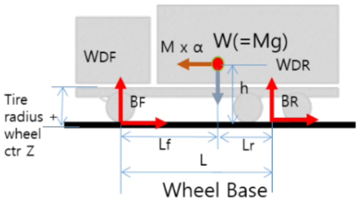


Figure 17: Free diagram for braking load calculation

In collaboration with the leaf spring manufacturer, a reliable 2D FEM simulation was conducted based on information such as leaf spring detailed shape information and stiffness test values. And it was confirmed that the optimal knuckle arm HP for large trucks 6x4 and 4x2 vehicles to use together is a position that has been moved 30mm in the vertical direction of the axle seat surface than the initial knuckle arm HP (see Figure 18)

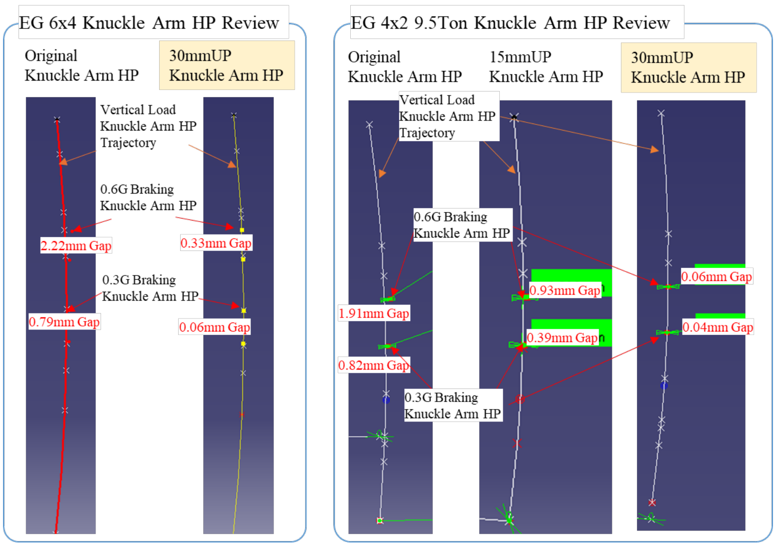


Figure 18: Finding the optimal knuckle Arm HP

In the end, a combination of design parameters using both the pitman arm and the knuckle arm in common and only the draglink in different was obtained and for the 6x4 dump truck, the change in the tire steering angle during braking was expected to be reduced to 0.17 to 0.41 degrees as shown in Table 4 below.

		6x4 Truck		4x2 Truck	
		Origin	mod	Origin	mod
Knuckle Arm HP		A	A'(30mmUP)	A	A'(30mmUP)
Length of draglink		B	B'	B'	B' - 65mm
PitmanArm Upr HP		C	E	D	E
Tire steer angle change	GVW to 0.3G braking	reference	0.17° reduced	reference	0.03° reduced
	GVW to 0.6G braking	reference	0.41° reduced	reference	0.10° reduced

Table 4: Large truck 6x4 / 4x2 design parameter combinations

A braking pull test was carried out to compare with the initial specifications of the 6x4 dump truck, and it was confirmed that the pulling feel of steering wheel is reduced in braking with holding steering wheel as shown in Fig. 19 below. and the modification have been reflected in the vehicle.

	Test Vehicle (6x4 Dump Truck)	
	Original Knuckle Arm HP origin Pitman Arm upper HP origin	**Modification** Knuckle Arm HP 30mm UP Pitman Arm upper HP 30mm UP
Braking pull	Satisfying the criteria	Satisfying the criteria
Pulling feel (braking)	RH turn pulling feel **1.0** (reference)	RH turn pulling feel **0.3**

Figure 19: the result of 6x4 large truck Braking pull improvement test

7. KINEMATIC REVIEW PROCESS IN THE EARLY DESIGN STAGE

In this study, the main design factors that influence the reliability of the kinetic review by design stage were identified. The main cause of the error in the Kinetic review results compared to the trend of the actual vehicle was the inaccuracy of the hard point information of the steering & suspension system in the loading state of the vehicle.

Since the initial hard point information of the steering and suspension system must be accurately predicted based on the leaf spring deformation according to the vehicle load, a kinematic review is required in consideration of the error range of vehicle weight, COG position, leaf spring camber, leaf spring stiffness, etc. in the early stages of design.

The accuracy of the relevant design information for each design stage can increase the accuracy of the review before vehicle production based on single-product measurement data. Therefore, for such a review, it was confirmed that although an analytical technique is also important, it is necessary to manage the information well for each design stage.

Finally, the steering Kinetic review process considering the leaf spring wind-up behavior during vehicle braking in the design stage can be summarized as shown in Figure 20 below.

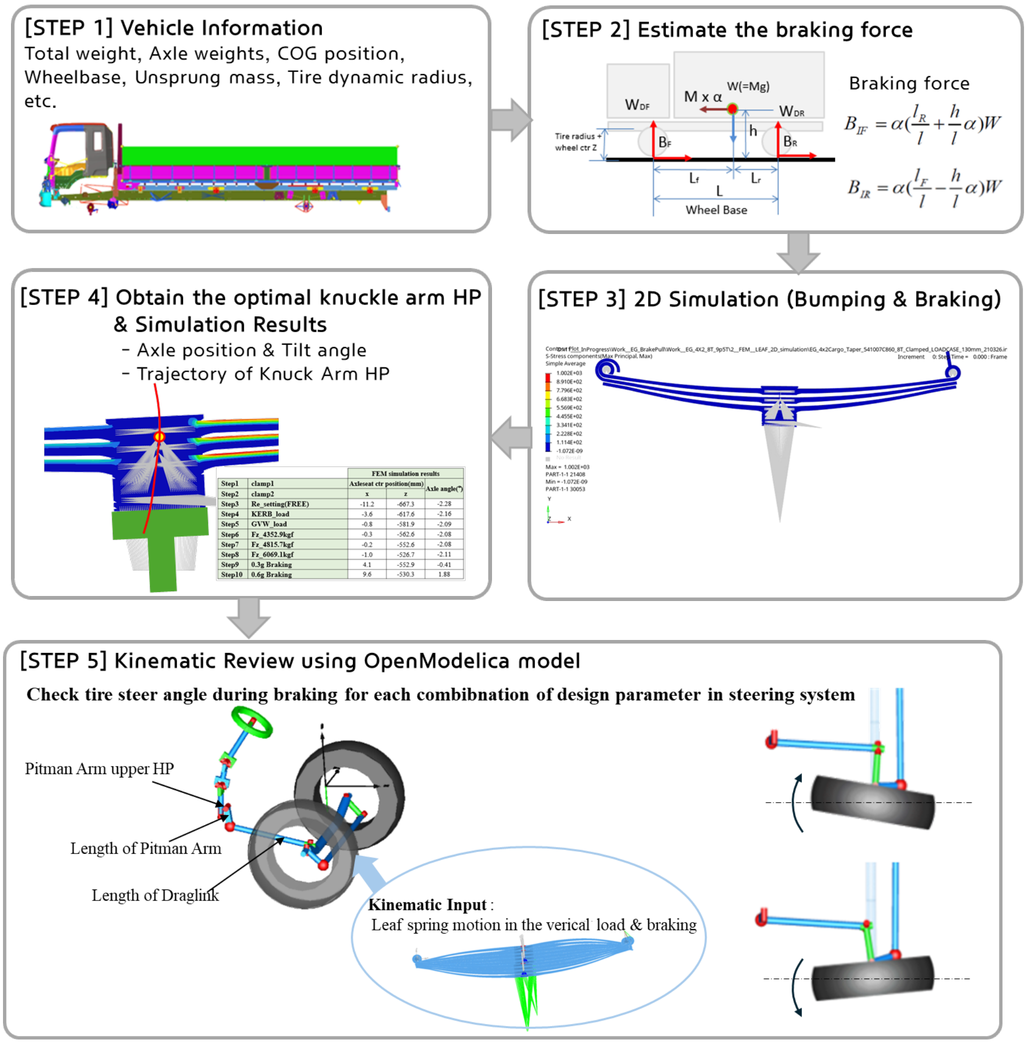


Figure 20: The process of kinematic review in design stage using 2D simulation

8. CONCLUSIONS

In this study, we developed a simulation methodology that performs a kinematic review considering not only vertical load but also leaf spring wind-up behavior during braking in the design stage of the leaf spring suspension system and steering system and established the process.

In particular, the reliability of the FEM 2D leaf spring analysis was confirmed by comparing the test values for the stiffness of the leaf spring and the windup behavior during braking, and if the shape of the leaf spring is manufactured at the design information level in the future, the geometric error review in the early stages of the design will be a meaningful review.

When reviewing geometry errors to minimize braking pull, the main experiences are as follows:

(1) In order to minimize braking pull by improving geometry errors, not only geometry errors during vertical loads but also geometry errors due to windup during braking loads should be considered.

(2) Before considering changes in pitman arm length and drag link length when reviewing geometry errors to improve braking concentration, it is necessary to prioritize the optimal knuckle arm HP position where the knuckle arm trajectory in vertical load and the knuckle arm HP positions in braking load are as close as possible.

(3) Once the optimal knuckle arm HP is determined, we find a combination of pitman arm length and drag link length that can reduce geometry error. However, if the pitman arm length and draglink length changes are expected to increase the minimum turning radius of vehicle and steering effect, the performance should be reviewed again to ensure that the performance meets the design criteria.

(4) Despite the above review, if it is difficult to obtain a satisfactory optimization design, you can try to find the optimal solution again by changing the leaf spring settings, such as changing the hanger bracket HP position and shackle length.

9. References

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