OPTIMIZATION OF SUSPENSION PARAMETERS BASED ON VEHICLE ROLL STABILITY AND ROAD FRIENDLINESS

Jingmei ZHANG[1,2*], Jianling SHI[2]

ABSTRACT: In this paper, three axle heavy duty vehicles are taken as an example. The rolling model of vehicles under random excitation is established. It is pointed out that due to the random excitation of the road surface and the steady motion of the roll, the vertical reaction force on the left and right sides of the tire changes. The vertical force of the inner tire is reduced, the outside is increased, and the tire cornering characteristics and the steady state response are affected. The influence of the stiffness and damping of the automobile steering and balance suspension on the steady state response of the vehicle is analyzed by using the vehicle yaw rate gain as the evaluation index. At the same time, it is pointed out that because of the change of the vertical load on the left and right sides of the tire, the road friendly performance of the road becomes worse, and the relationship between the parameters of the suspension is analyzed by the evaluation index: the sum of four times power of force at the 95 percentile. And the constraint conditions are set up. Based on the improved non dominated sorting genetic algorithm (NSGA-II), the suspension parameters are optimized and matched, so that the stability and the road friendliness of the vehicle can reach the desired state when the car is on the roll.

KEYWORD: Roll stability, road friendliness, steady yaw angular velocity gain, optimization

1. INTRODUCTION

With the rapid development of Chinese economy, using the technology of truck is increasingly mature, automotive products increasingly fierce competition, heavy truck technology in China has made progress in leaps and bounds, but relative to developed countries is still in the initial stage, some of the technical problems have to be solved. Heavy haul vehicle accidents happen frequently, especially for overloaded cars. Cornering accidents during cornering are common, causing heavy losses to people's lives and property.

Corresponding author: Jingmei Zhang
[1]School of Mechanical, Electric and Control Engineering, Beijing Jiaotong University, Beijing, China
[2]School of Mechanical Engineering, Hebei University of Science and Technology, Shijiazhuang, China
E-mail: zhang_jiangmeijm@163.com

At the same time, because of the overloading of vehicles, the damage to the pavement and the extent of the broken ring increase (Maghsood et al, 2005), the safety factor of the road decreases, and in turn affects the driving of the vehicle. In recent years, many scholars have studied vehicle dynamics for cornering vehicles for a long time, but mainly on vehicle performance (Yuming Yin et al, 2016).Therefore, the mutual influence theory of the handling stability and road friendliness of the freight car when cornering is one of the major issues worth studying.

A rolling model of three axle heavy haul truck turning at a constant speed established in this paper. The road random excitation is constructed three incentive model with the time lag. The relationship both the vehicle roll stability evaluation index: yaw velocity gain and suspension stiffness, damping is obtained. Combined with road friendliness damage coefficient the simulation analysis is carries on.

2. ROLL MODEL AND STEADY STATE RESPONSE

The vertical load on both sides of wheels is basically the same when the car runs in a normal straight line. But in the actual running, the traffic route changes constantly, often appears the curve traveling condition. It is assumed that the traveling speed is the uniform. The automobile produces the centrifugal force and the lateral tilt when the automobile moves along the curve. The roll moment causes the bilateral tire vertical load to be no longer the same. The automobile steady state response also has changed. In the whole body roll, front and rear suspension system will produce lateral deformation which changes the tire cornering stiffness, lateral force and the elastic slip angle of tire so that vehicle stability and comfort is greatly affected. Figure 1 is a schematic diagram of a three axle truck when it rolls. F_{sy} is centrifugal force.

Figure 2 shows the force diagram of a single bridge under roll. The roll characteristic of suspension mainly refers to camber stiffness of suspension. The restorative force of the suspension system when the car body is doing vertical displacement is the spring force and the damping force for non-independent

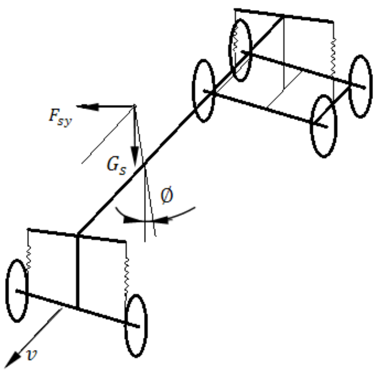

Figure1. Roll scheme

suspension systems. This resilience is actually the difference between the reverse force of the road and the non-suspended mass. It is very easy to find the roll angle stiffness of the suspension:

$$K_{\beta i} = (K_i + \frac{c\dot{z}_i}{z_i})B'^2 i \quad i = 1,\ 2, \qquad (1)$$

Where K_i is the suspension spring stiffness, C_i is the spring damping, Z_i is the spring mass vertical displacement, B' is the spacing between the sides of the spring. The subscript: $i = 1, 2, 3$ represents the front, the middle and the rear axle respectively. The following are the same as.

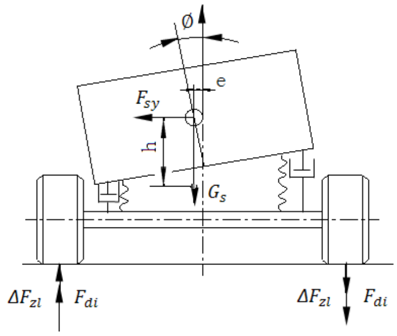

Figure2. A diagram of the force on the side of the vehicle

The vertical reaction of the left and right wheels is discussed only by the side tilting moment and the vertical moving load except to the equilibrium forces of the gravity and the ground reaction of each wheel. Let's say the body is tilted to the left, the load on the three shafts can be solved.

(1) The vertical load of the left and right wheels when the car rolls

The front axle is simplified equivalently by isolation method, and the load distribution diagram is shown in figure 3. The change in the vertical movement load of each tire is due to the amount of the change in the dynamic load of the side and the dynamic load generated by road stimulation.

At this point, the vertical load on the three shafts is no longer the same as the straight line, at the same time, the random excitation of the road makes the tire force change, resulting in a vertical dynamic load, so the load on the three axle left and right tires is given by respectively:

$$F'_{zil(r)} = F_{zil(r)} + \Delta F_{zil(r)} + F_{id(r)} \qquad (2)$$

Where, F_{id} is the dynamic load generated by the road excitation on the three axis, and $F_{id} = M_l \ddot{Z}_l$ ($i = 1, 2, 3$), L is left and r is right. They have the same meanings hereinafter.

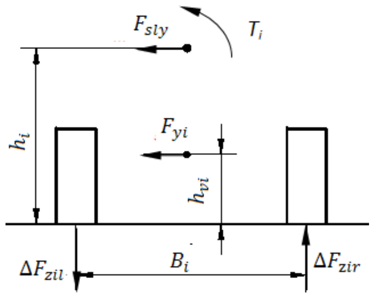

Figure 3.The schematic diagram of three axis lateral reaction force

$F_{zil(r)}$ is the vertical reaction force of the left (right) tire of the front, middle, and rear axle.

The dynamic load of the inner tire produced by roll can be expressed as:

$$\Delta F_{zil} = (F_{si} h_i + T_i + F_{vi} + F_{vi} h_{vi}) / B_i \quad (3)$$

$$K'_{il(r)} = 0.06778 F'^{\,2}_{zi(r)} - 9.144 F'_{zi(r)} + 5.129 \quad (6)$$

Thus, the improved steady-state factor formula is given by:

$$K_{l(r)} = \frac{mv^2}{L^2}\left(\frac{a}{k_{1l}} - \frac{L_2}{k_{2l}} - \frac{L_1 + L_2}{k_{3l}}\right) \quad (7)$$

When the stability factor of the tire on the left and right sides changes, the steady state yaw rate gain of the vehicle can be expressed as:

$$\gamma = \left[\frac{(u / l)}{1 + K_l v^2}\right] + \left[\frac{(u / l)}{1 + K_r v^2}\right] \quad (8)$$

$$\Delta F_{zir} = -\Delta F_{zil} \quad (4)$$

Where F_{si} is the distribution values of centrifugal forces: $F_{sy} = m_b \frac{V^2}{R}$ on three axes. T_i is roll moment of each axle. And:

$$T_i = K_{\beta i} \beta \quad (5)$$

Where

$$\beta = (M_s \frac{v^2}{R} + M_s g e) / [(K_1 + C_1 \dot{Z}_1 / Z_1) B'^{2}_f + (K_2 + C_2 \dot{Z}_2 / Z_2) B'^{2}_2 (K_3 + C_3 \dot{Z}_3 / Z_3) B'^{2}_3]$$

By substituting Eq.(4) and Eq.(6) into Eq.(3), the inner and outer tire reaction forces can be obtained when each axle rolls.

(3) Steady state response of vehicle

For a constant speed vehicle, the steady-state response of the vehicle is evaluated by the ratio (the yaw rate gain) of the steady-state yaw rate and the front wheel angle. According to the experiment and Eq.(2), the cornering stiffness of the tire is obtained:

(4) The influence of suspension parameters on steady state response

Steady state yaw angle speed gain is the main index to measure the steady running of vehicle. The ground reaction force of the tire has changed because of the lateral force and the random excitation of the road surface when the roll, i.e. wheel vertical load has changed, and the tire cornering stiffness is influenced. This also affects the steady-state yaw speed gain of the vehicle. Figure 4 to Figure 5 show the influence relationship between the stiffness and damping of the front and rear suspension on the steady-state yaw rate gain of the vehicle.

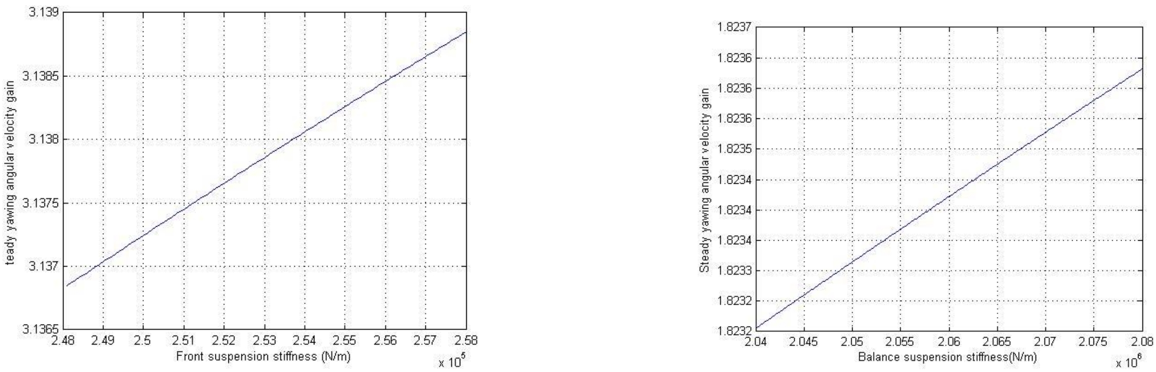

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Figure4. Relationship between steady-state yaw speed gain and front suspension or equilibrium suspension stiffness

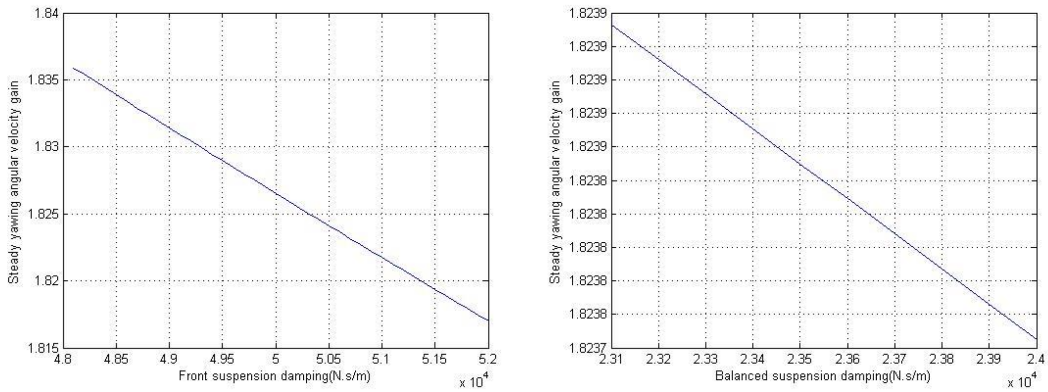

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Figure5. Relationship between steady-state yaw rate gain and front suspension or equilibrium suspension damping

As shown by the diagram, the steady-state yaw rate gain of the vehicle increases with the increase of the suspension stiffness and decreases with the decrease of the suspension damping.

3. THE EVALUATION INDEX OF ROAD FRIENDLINESS

The sum of four times power of force at the 95 percentile of the front axle, the middle axle and the rear axle of the three axle heavy load vehicle are added by the evaluation index of road friendliness(D J Cole et al 1996;Potter T E C et al,1997) flowing as :

$$\begin{aligned}\phi &= (\eta_{11}\eta_{21}\eta_{31}F_{1-stat})^4 + (\eta_{12}\eta_{22}\eta_{32}F_{2-stat})^4 + (\eta_{13}\eta_{23}\eta_{33}F_{3-stat})^4 \\ &= [1.1\times(1+1.645DLC_1)F_1]^4 \\ &+[0.9\times1.1\times(1+1.645DLC_2)F_2]^4 \\ &+[0.9\times1.1\times(1+1.645DLC_3)F_3]^4\end{aligned} \quad (9)$$

η_{1i} is influence coefficient of front, middle and rear tire arrangement. Because the rear axle is double axis in the paper, η_{11} is equals to 1.0, both η_{12} and η_{13} equal to 0.9). In addition, the influence coefficient η_{2i} of tire inflation pressure is relating to tire pressure. According to this example, the tire pressure of the actual vehicle tires (11.R20) is equals to about 0.88MPa, so η_{2i} is equals to 1.1 (Arbeitsausschuss, 1978).

When the car rolls, the load on the left and right tires changes, and the dynamic load coefficient also changes, which affects the sum of four times power of force at the 95 percentile. The dynamic load on the left tire of the vehicle is given by:

$$F_{dil} = \Delta F_{zil} + M_l \dot{Z}_l \quad (10)$$

The frequency response function of the left tire dynamic load relative to the ground excitation input can be obtained by the virtual excitation method (J.H. Lin et al, 2011):

$$H(f)_{Fidl-q1} = F_{dil} / \tilde{q}_1$$

$$= \Delta F_{zil} A e^{jwt} - M_1 \omega^2 H_{zi-q1} \quad (11)$$

A presents self-power spectral density for road excitation.

The input excitation is filtered white noise, and the power spectrum density function of the left tire dynamic power can be written as:

$$G_{Fidl}(f) = \left| \Delta F_{zil} A e^{jwt} - M_l \omega^2 H_{zi-q1} \right|^2 G_{q1(f)}$$

$$= G_{q1}(n_0) n_0^2 v \left| \Delta F_{zil} A e^{jwt} - M_l \omega^2 H_{zi-q1} \right|^2 / f^2 \quad (12)$$

Thus, the dynamic load coefficient of the front axle generated by the revolver tire is given by:

$$DLC_{il} = \sqrt{\int_0^\infty (G_{idl}(f))df} / F_{i-atat} \quad (13)$$

The sum of four times power of force at the 95 percentile produced by the left tire force can be written as:

$$\phi_l = [1.1 \times (1 + 1.645 DLC_{1l}) F_1]^4$$

$$+ [0.9 \times 1.1 \times (1 + 1.645 DLC_{2l}) F_2]^4$$

$$+ [0.9 \times 1.1 \times (1 + 1.645 DLC_{3l}) F_3]^4 \quad (14)$$

By Eq.(13) and Eq. (14), the sum of four times power of force at the 95 percentile of the left tire can be obtained. Similarly, the sum of four times power of force at the 95 percentile of the right tire can be obtained. Then the sum of four times power of force at the 95 percentile of the whole vehicle is equals to the left's and the right's.

4. OPTIMAL MATCHING OF SUSPENSION PARAMETERS

It is assuming that the car is running conditions on B class road, turning at uniform speed of 30km/h.

(1) Optimization objective

The paper takes the road friendliness evaluation index: the sum of four times power of force at the 95 percentile and steady-state response index of roll stability: steady state factor K as the optimization objectives. In order to take into account the roll stability and road friendliness of automobiles, It should be kept that road damage from the vehicle is minimized up and the steady-state yaw rate gain is moderate, .i.e. the steady-state factor K satisfies Max K, and k>0.

According to the reference (Byeong Seok Ahn, 2017), the weighted proportionality coefficient is determined by using analytic hierarchy process because the importance of roll stability and road friendliness should be considered. On the premise of driving safety, the roll stability is weighted to the road friendliness.

(2) Design variables and constraints

It is possible to know the stiffness of the front suspension and the damping of the damper, the stiffness and damping of the equilibrium suspension are closely related to the ride comfort of the vehicle according to the analysis of front roll stability and road friendliness. So the stiffness and damping of the front and rear suspension is chosen as the design variables for optimization analysis:

$$X = \left| K_r \; C_r K_f C_f \right|^T \quad (15)$$

Where K_r or C_c is the balance suspension stiffness and damping respectively; K_f *or* C_f is the front suspension stiffness and damping respectively.

The range of stiffness optimization is determined according to the frequency of heavy truck, the front suspension load frequency range is 1. 50~2. 10Hz, the full frequency range is 1.7 ~2. 17hz (J.-D. Müller et al, 2012).The damping coefficient of suspension is generally chosen

$C_r = 2\xi\sqrt{K_r M_r}$,In the formula, ξ is relative damping coefficient. For friction leaf springs, the value of ξ is a little smaller, Optional according to practice and experience $0 \le \xi \le 0.2$.

(3) Optimization platform

The suspension parameter optimization of roll stability and road friendliness of truck is carried out by using optimization software I sight. The software provides optimization algorithms, such as optimization software package, easy to implement integration with other software. Select improved non-dominated sorting genetic algorithm NSGA-II, integrated Matlab components and Import rollover stability and road friendliness form files, and set up the mapping relationship and constraints of the input and output parameters.

5. OPTIMIZATION RESULTS AND ANALYSIS

The Pareto optimal solution set of the design variable is obtained by optimization. The mapping of Pareto optimal solution set in objective function space, that is, Pareto is shown in figure6 to figure 7.

From Figure 6 and Figure 7, the damping and stiffness of the steering suspension is larger for Pareto optimal set mapping when road friendliness in the optimal state, and the damping and stiffness of the balanced suspension is small for the Pareto optimal solution set mapping.

In the optimization, the amount of dynamic deflection of the suspension should be ensured as:

$f_d = 60 \sim 90mm$.

Figure 7 indication optimal solution set mapping is the larger damping and stiffness of the steering suspension when the steady-state gain is in the optimal state, and it is the smaller damping and stiffness of the balanced suspension.

To sum up, the optimization of vehicle roll stability is carried out after considering the friendliness of road. Table 1 shows the data before and after the optimization of the design variables:

The data in Table 1 shows that the suspension parameters have changed after optimization, the stiffness of the balanced suspension is 19.82% lower than that before optimization and the equilibrium suspension damping ratio is reduced by 8.59% before optimization.

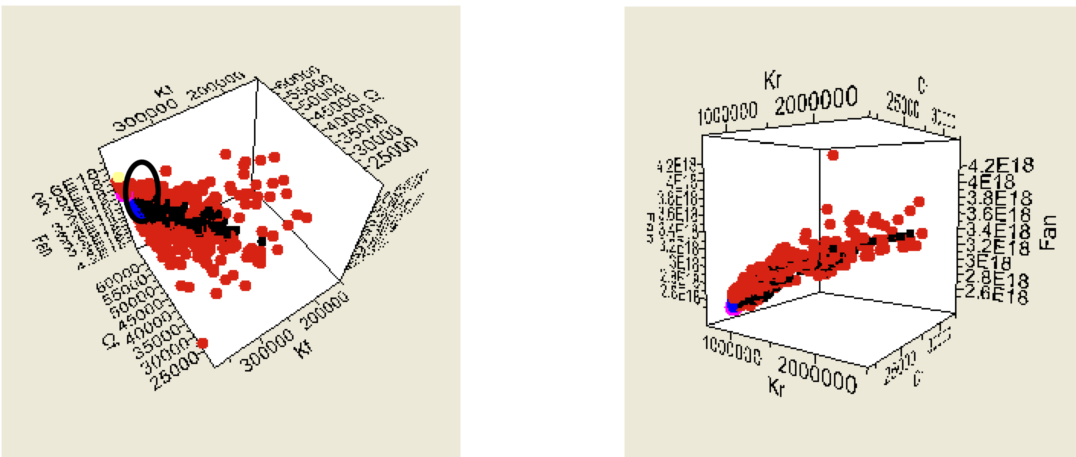

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Fig. 6 Pareto optimal solution set of front suspension or equilibrium suspension stiffness and damping (for road friendliness index)

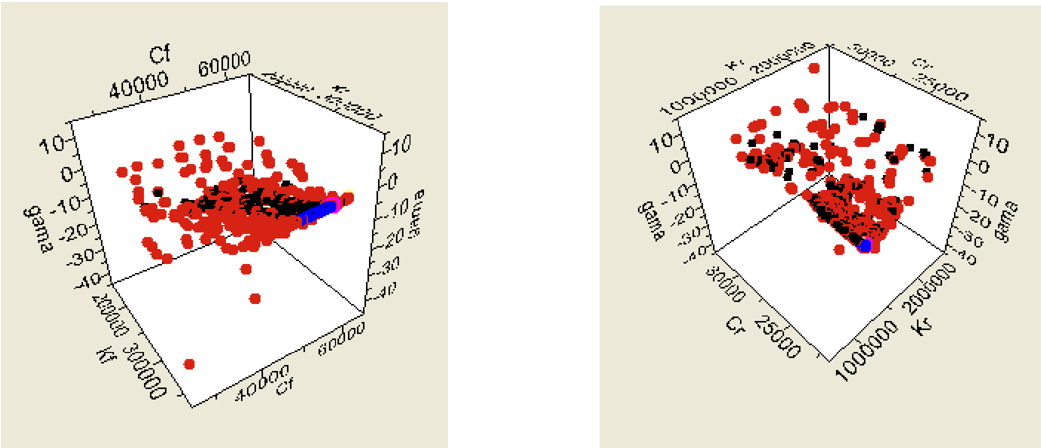

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Fig. 7 Pareto optimal solution set of front suspension or equilibrium suspension stiffness and damping (for the roll stability)

Table 1 Comparison of optimization results

Optimization results	K_r (N/m)	C_r (N▪s/m)	K_f (N/m)	K_f (N▪s/m)
Before optimization	2304600	32600	370000	60000
After optimization	1047550	21800	168000	25000

The front suspension damping ratio is reduced by 8.33% before optimization and the stiffness of the front suspension is improved by 5% before the optimization.

Under the matching of the optimized suspension parameters in the table, the roll stability of the vehicle is at its optimum when the road friendliness is the best. The result of the optimized objective function is shown in Table 2.

It can be seen from table 2 that the steady-state yaw rate increased by about 16.78%, Sum of four times power of force at the 95 percentile is reduced by about 12.6% after the front and rear suspension stiffness

Table 2. Comparison of optimization results of the objective function

Objective function	Steady-state yaw rate gain	Sum of four times power of force at the 95 percentile($10^{16}N^4$)
Before optimization	0.2675	2.2832
After optimization	0.3124	1.9959

and damping optimization are matched. So that the vehicle roll stability and the road friendliness were both improved. The whole process of the optimization ideas provides some important reference for the actual car related engineering design.

6. CONLUSION

This paper mainly concerns the related parameters of vehicle, and analyzes and optimizes the rolling stability based on friendly Road. The main work is summarized as follows:

1) The steady-state yaw angle gain of three axle heavy vehicle is Improved When calculating the roll moment. In the calculation of roll torque, suspension stiffness and suspension damping are considered. Improved the steady-state yaw rate gain is obtained. A quantitative relationship between the steady-state amplitude gain and suspension stiffness and damping is established. It is found that the steady-state yaw velocity increases with the increase of the rigidity of the suspension, Then the stability becomes worse. It is also known than the steady-state yaw velocity decreases with the decrease of the damping of the suspension, and the stability becomes worse.

2) It is put forward that the front and rear suspension stiffness and damping is taken as the design parameters of optimization analysis and the evaluation indexes of road friendliness into four power and force and yaw velocity gain optimization: sum of four times power of force at the 95 percentile and yaw velocity gain are taken as the optimization objective. The suspension stiffness and damping was optimized. Through the comparison the optimization objective function before and after, we can get the best matching parameters of the front and rear suspension.

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