Defence Science Journal, Vol. 60, No. 2, March 2010, pp. 178-183

Wishbone Structure for Front Independent Suspension of a Military Truck

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ABSTRACT

Wishbone structure for double wishbone front-independent Suspension for a military truck application is presented. At present, the vehicle is equipped with rigid axle with leaf springs. There are two aspects that dictate the design of wishbone structure, viz. the path of relative motion between the constituents of the suspension system and the forces transmitted between them. Also, enhancement of mobility was made possible by maintaining the live axle in the system. A double wishbone, double coil spring with twin damper configuration was employed for this application. MBD Analysis was carried out using MSC ADAMS. A double wishbone-independent suspension has been designed for the front axle and has been successfully integrated with the vehicle.

Keywords: Independent suspension, live axle, double wishbone configuration, MBD analysis

NOMENCLATURE

d	Offset of upper arm
e	Offset of lower arm
H	Distance of upper ball joint from ground
h	Distance between upper ball joint and lower ball joint
Z	Lift of the wheel centre
Z_1	Lift of upper arm
Z_2	Lift of lower arm
Y_1	Lateral displacement of upper ball joint
Y_2	Lateral displacement of lower ball joint
R_1	Length of upper arm
R_2	Length of lower arm
a	Vertical distance of upper ball joint wrt inboard point
b	Vertical distance of lower ball joint wrt inboard point

1. INTRODUCTION

Independent suspension systems can be provided by a variety of linkages between the stub axle carrying the wheel and the vehicle chassis. Most popular combinations in modern passenger cars are MacPherson struts for steered wheels and semi-trailing links for un-steered rear wheels. The double wishbone configurations can give vertical movement close to perpendicular relative to the tyre contact surface. The classic twin, unequal wishbone linkages arrangement provides lower unsprung mass and a wide selection of spring types can be considered to give very good wheel adhesion and optimum wheel control. Independent suspension system provides a whole new range of possibilities in regard to the application of computer systems which would provide active suspension control to suit various conditions of load, speed, and terrain in vehicle operation. Application of independent suspension systems to military trucks presents various problems that may not be immediately evident. The scale effects are not linear, so increasing the linear dimensions pro rata is not a valid approach. Military trucks, in general tend to use live axle suspension systems with rigid axles. These systems provide limited wheel travel and indefinite control of wheel movement and location. When high performance travel on rough terrain is required, additional wheel travel is provided by allowing the flexible chassis to twist substantially. Independent suspension system has not been used with trucks so far in spite of increased driver comfort and better vehicle performance.

There is no single best geometry for the independent suspension system of a vehicle. Any particular geometry must be designed to meet the design needs of the vehicle to which it is applied.

2. DOUBLE WISHBONE STRUCTURE

Figure 1 shows a typical configuration of double wishbone independent suspension. It consists of upper and the lower wishbone arms pivoted to the frame member. The spring is placed in between the lower wishbone and the underside of the cross-member. The vehicle weight is transmitted from the body and the cross-member to the coil spring through which it goes to the wishbone member. A shock absorber is placed inside the coil spring and is attached to the cross-member and to lower wishbone member. The wishbones not only position the wheels and transmit the vehicle load to the springs, but also resist acceleration, braking, and cornering forces. The upper arms are shorter in length than the lower ones. This helps to keep the wheel track constant, thereby avoiding the tyre scrub, thus minimising

Received 4 March 2009

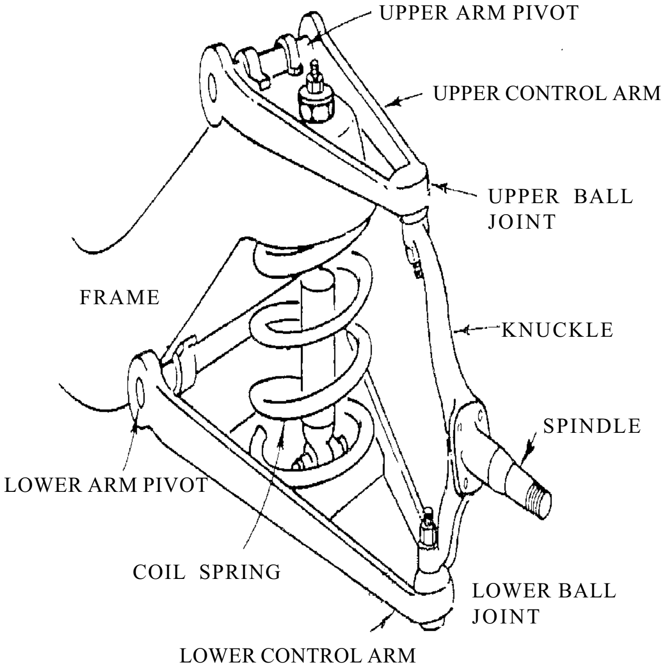

Figure 1. General double wishbone configuration.

tyre wear. However, a small change in the camber angle does occur with such an arrangement.Thus, the geometry of double wishbone structure plays an important role in determining tyre life, more comfort and better ride handling.

3. RIGIDLY HELD DIFFERENTIAL HOUSING

No clearance relative to the chassis or its load is needed for the differential bowl movement as there is no movement of the differential housing with this configuration. Fig. 2 shows the layout of the independent suspension system with rigidly held differential housing.

4. CONTROL ARM GEOMETRY

The control arm geometry was selected such that the track width of the original vehicle remains unchanged. The ground clearance which is an important criterion for a military vehicle was increased from 305 mm to 320 mm. Different possibilities of configurations, viz., coil springs mounted on lower control arm, additional link between steering knuckle and upper control arm, and coil springs mounted on upper control arm were studied. Double coil spring with twin damper configuration was used. The loads coming on to the control arms were high. The availability of space for the steering linkages imposed a constraint hence the spring damper arrangement was mounted on the upper control arm itself. Initially it was planned to use as many parts of the original vehicle however, since the steering linkages were in front, the steering knuckle had to be re-designed. The wheel travel with this configuration is 150 mm in bump and 100 mm in rebound, which is substantially higher than the original vehicle. Figure 3 shows the nomenclature used in wishbone design.

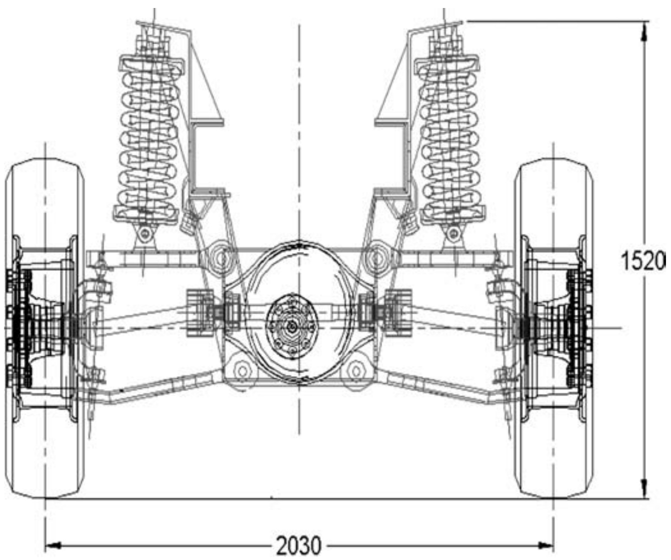

Figure 2. Layout of the system.

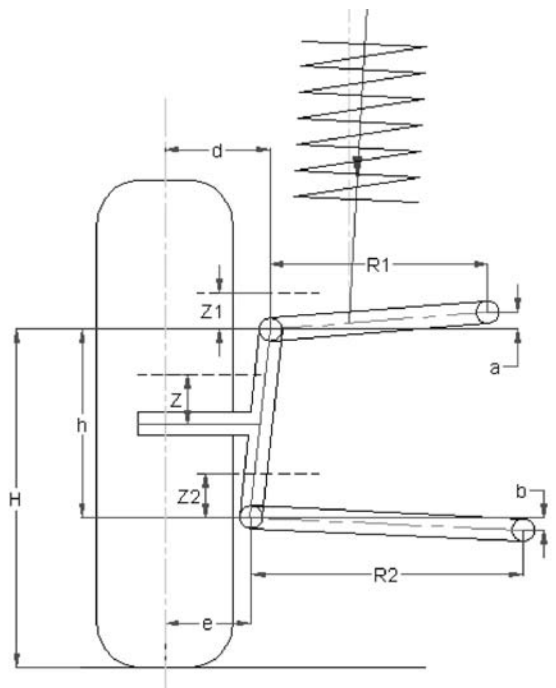

Figure 3. Notations used in wishbone design.

d = 234 mm

e = 190 mm.

H = 747.5 mm

h = 420 mm

Z = 180 mm

For deciding the length of control arms, the travel of the wheel in upward and downward motions (bump and rebound) was considered. The length of control arms was calculated using the following equations

$$Y_1 = \frac{(Z_1)^2}{2 \times R_1} + \frac{(Z_1 \times a)}{R_1} \tag{1}$$

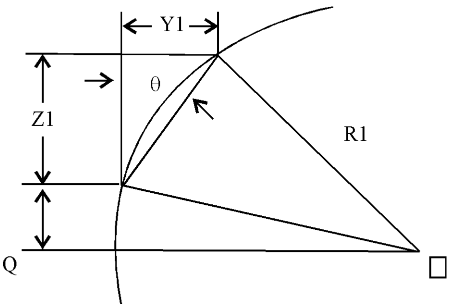

Figure 4. Control arm path from static to bump.

$$Y_2 = \frac{(Z_2)^2}{2\times R_2} + \frac{(Z_2\times b)}{R_2} \tag{2}$$

Figure 4 shows control arm path from static to bump.

For triangle PQR for given value of *q,* i.e., allowable change in camber angle say 5^0. And required bounce Z = 180 mm, we can find the distance Y_2.

$$Z_2 = Z - e\times\theta = 180 - 190\times\left(5\times\frac{\pi}{180}\right)$$

$$= 163.4 \text{ mm.}$$

$$\tan\theta = \frac{Y_2}{Z_2}$$

$$Y_2 = \tan\theta\times Z_2 = \tan 5\times 163.4$$

$$= 14.29 \text{ mm}$$

Similarly, Z_1 = 160 mm

Y_1 = 13.99 mm

Now puting values of Z_2 and Y_2 in Eqn 1

$$14.29 = \frac{(163.4)^2}{2\times R_2} + \frac{163.4\times(-29)}{R_2}$$

$$R_2 = 602.60 \text{ mm}$$

Similarly,

$$13.99 = \frac{(160)^2}{2\times R_1} + \frac{160\times(-38)}{R_1}$$

$$\therefore \quad R_1 = 477.95 \text{ mm}$$

Figure 5 shows the final dimensions of the wishbone configuration. The variation of control arm length wrt camber angle was plotted and the length of control arms was finalised.

Length of upper control arm = 480 mm

Length of lower control arm = 602 mm

5. MECHANICS OF THE INDEPENDENT SUSPENSION SYSTEM

Graphical method was used for the determination of the forces acting at various points in the suspension system. Fig. 6 shows the force analysis of the wishbone configuration.

Axle and wheel assembly in equilibrium under *R* and force acting on the pivots *A* and *C*. There is no transverse force acting on link *AB*. Therefore, force acting on pivot A must be along *AB* giving intersection of its line of action with that of R at the point *G* through which the force *U* at pivot *C* also passes. The force *P* on link *AB* is tensile and force *Q* on link *CD* is compressive. The forces in spring denoted by *S* and a, ß are respectively inclinations of links *AB* and *CD* with horizontal and γ denotes the inclination of *SE* to vertical.

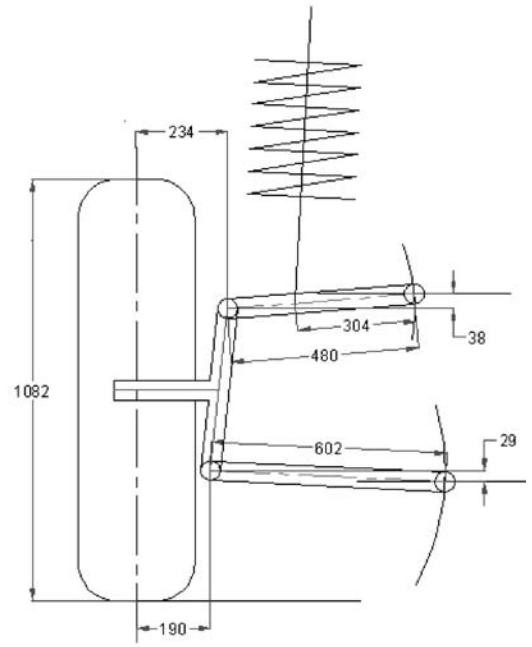

Figure 5. Final dimensions of the wishbone configuration.

Input:

Reaction on front axle = 5675 kg

Reaction at one wheel = $\frac{5675}{2}$ = 2837.5 kg = 27835.9 N

AB = 480 mm

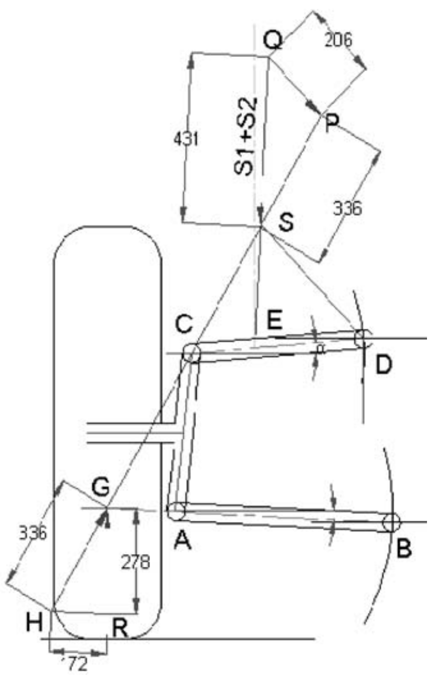

Figure 6. Force analysis of wishbone structure.

CD = 600 mm
CE = 175 mm
DE = 305 mm
α = 3 °
β = 4 °

5.1 Procedure for Static Force Analysis

Step 1. A force scale of 100 N = 1mm was established. *A* vertical line is drawn at wheel centerline.

Step 2. For equilibrium the lines of action of the forces at the upper and lower ball joints at the wheel centerline must all cross the point *G*. This is equivalent to analytical condition, $\sum M = 0$

Step 3. A line is drawn from point *B* to point *A* and it intersects the centerline at point *G*. The force acting at the wheel centerline (i.e. 27836 N = 278 mm) is drawn at point *R* from point *G*.

Step 4. A line is drawn from point *G* to point *C*. This line is extended so that it intersects the line parallel to line *GB* at point *H*.

Step 5. From this, the force at point *C* (upper ball joint) from line *GH* is obtained, i.e., 33600 N.

Step 6. The line *GC* is extended so that it intersects the line of action of spring force at point *S* and an arc of 336 mm is drawn.

Step 7. A line is drawn from point *D* to point *S* and a line *PQ* parallel to *DS* is drawn at *P* which intersects the line of action of spring force at point *Q*.

Step 8. Thus, spring force is obtained from the diagram (*QS*) and is found to be 43100 N. Spring was designed for this force.

5.2 Computer Simulation and Analysis

A software package MSC ADAMS was used for carrying out multi-body simulation of the system.

5.2.1 Wheel Travel Analysis

Wheel orientation such as camber, track change between extreme wheel travel positions and steering positions were predicted and compared to the objectives. The analysis of the suspension was carried out for all the possible combinations of steering and wheel travel. It was also checked for clash between components in the suspension for all the possible combinations. Figure 7 shows one of the possible combinations, i.e., opposite wheel travel analysis.

5.2.2 Road Load Data Simulation

The vehicle was mounted with accelerometers near the wheel centres. The accelerations were measured, while displacements needed to be provided as input, made it necessary to double-integrate the road signals and thus preventing the direct feeding of road signals into the system. This problem would have been simplified if wheel displacements could have been recorded directly. In this case, no double integration would have been needed. However, accelerometers are much easier to handle wrt a definite zero reference signal, and wrt low frequency response, than the conventional displacement transducer. Using this input, loads acting on the suspension system components, viz. the wishbone structure, steering knuckle were obtained. Fig. 8 shows forces obtained on one the suspension components.

5.2.3 Finite Element Analysis

The loads obtained from the analysis were used for applying boundary conditions. The solid model was meshed

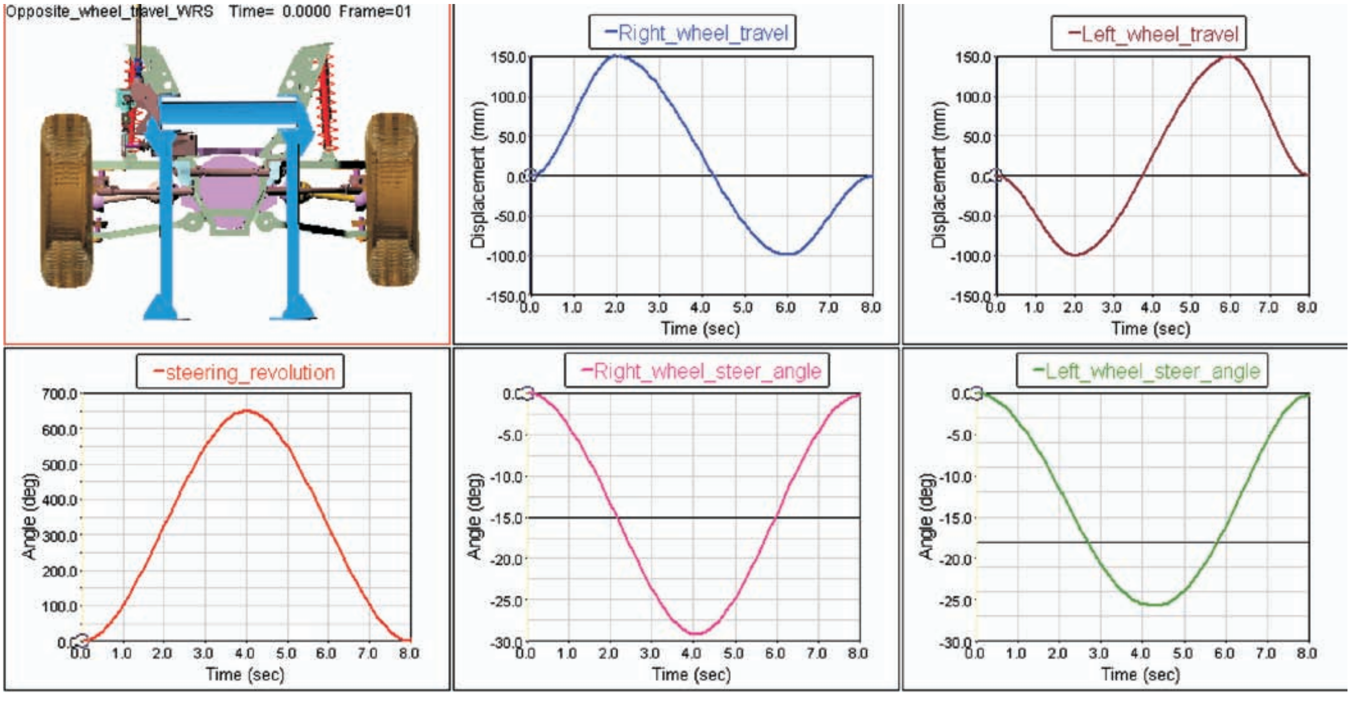

Figure 7. Opposite wheel travel analysis.

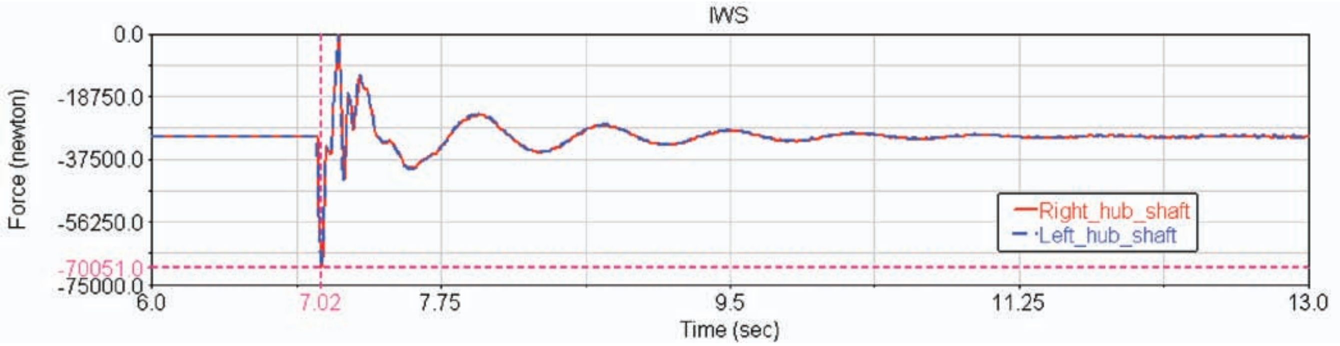

Figure 8. Suspension forces acting on one of the components.

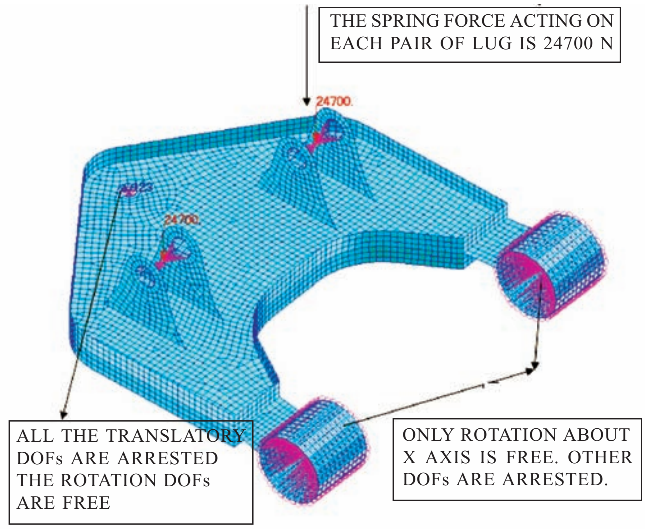

Figure 9. Finite element analysis of upper control arm.

Figure 10. Independent suspension system integrated with the vehicle.

using HyperMesh software and was solved using Abaqus-Standard software. In the similar fashion FE analysis was carried for all the components viz., lower control arm, spring mount bracket, sub-frame, steering knuckle and the steering system components. Figure 9 shows the FE analysis of upper control arm.

5.3 Integration and Trials

Suspension geometry details are provided in Table 1. The system was integrated on the vehicle. The methodology adopted for the development involved various steps, viz.,

- Identification of design constraints
- Evolvement of preliminary configuration
- Analysis for loading
- FEA analysis for deflection and stress under various loading conditions and refinement of design
- Limited technical trials to assess the structural integrity and mobility aspects of the vehicle.

Table 1. Suspension geometry

Parameter	Value
Front track	2030 mm
Steering axis inclination	5°
Ground clearance	320 mm
Toe- in	Parallel
Camber angle	1.5°
Caster	0°

Figure 10 shows the independent suspension system integrated with the vehicle. The vehicle was run on pave and pitching tracks at National Centre for Automotive Testing (NCAT), VRDE. No visible defect or deformation was observed after trials.

6. FUTURE WORK AND OTHER POSSIBILITIES

This type of configuration lends to usage of wide variety of spring media. This work relates to coil springs. Future developments with active suspension system are likely to centre on air, or hydro-gas suspension systems, but electro-rheological fluid units may ultimately prove to be the most suited for this from of computer-controlled suspension. Possibility of using hydrogas struts in the system is being considered.

7. CONCLUSIONS

High performance on rough terrain can be provided using this configuration. In spite of the demand of increased driver comfort and better vehicle performance, so far, independent suspension system has not been used for military trucks. With rigidly held or dead differential housing, no clearance relative to the chassis or its load is needed for the differential bowl movement as there is no movement of the differential movement with this configuration. In case of new vehicle development with this configuration, the engine can be lowered reducing the CG height. The independent suspension system for a military truck has been developed and successfully integrated on the vehicle. The maximum safe turning speed at 24 m radius is 32 km/h. The ground clearance has been increased from 305 mm to 320 mm. The wheel travel also has been increased from 150 mm (approx) to 250 mm. Independent suspension system applied to military truck offers attractive advantages in regard to comfort and road holding.

ACKNOWLEDGMENTS

The authors are grateful to Dr C.L. Dhamejani, Director, VRDE, for granting permission for publishing this work. The authors are grateful to Shri V.K. Pandey, STA 'A' and staff of Development Workshop for contribution in the development work. The authors are also thankful to CADES Digitech Pvt Ltd, Bangalore, for providing their valuable support.

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