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Optimisation of Kinematics for Tracked Vehicle Hydro Gas Suspension System

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ABSTRACT

The modern-day armoured fighting vehicles (AFVs) are basically tracked vehicles equipped with hydro gas suspensions, in lieu of conventional mechanical suspensions like torsion bar and coil spring bogie suspensions. The uniqueness of hydro gas suspension is that it offers a nonlinear spring rate, which is very much required for the cross-country moveability of a tracked vehicle. The AFVs have to negotiate different cross-country terrains like sandy, rocky, riverbed, etc. and the road irregularities provide enumerable problems during dynamic loadings to the design of hydro gas suspension system. Optimising various design parameters demands innovative design methodologies to achieve better ride performance. Hence, a comprehensive kinematic analysis is needed. In this study, a methodology has been derived to optimise the kinematics of the suspension by reorienting the cylinder axis and optimising the load-transferring leverage factor so that the side thrust on the cylinder is minimised to a greater extent. The optimisation ultimately increases the life of the high-pressure and high-temperature piston seals, resulting in enhanced system life for better dependability.

Keywords: Optimisation, kinematics, hydro gas suspension, tracked vehicle, cross-country mobility, single slider crank mechanism, kinematic linkages

1. INTRODUCTION

Mobility is one of the important characteristics of a tracked vehicle and is determined mainly by the power of the engine and drive train. The running gear subsystem is a part of the drive train on which the vehicle is mounted and it is the main subsystem that provides mobility to the tracked vehicle. The suspension is one of the important constituents of the running gear, which is shown in Fig.1 for a typical tracked vehicle meant for armoured fighting vehicle (AFV) application.

The suspension takes the entire vehicle weight and offers a flexible support to the vehicle on the ground. The need of a hydro gas suspension system for the tracked vehicles arose due to its inherent advantages over the conventional suspension system. The hydro gas suspension is a state-of-the-art suspension system offering smoother ride quality with reduced crew fatigue. It not only improves the cross-country mobility but also significantly enhances the weapon system accuracy and reliability in case of AFV application as it provides a more stable gun platform compared to the conventional suspension systems. In the military vehicles application, the hydro gas suspension technology is considered to be a frontier technology to ensure a high-level performance of onboard system, at par with the level of development

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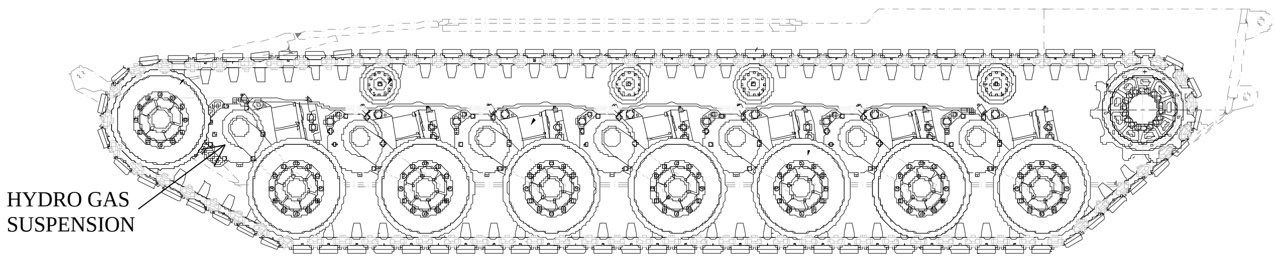

Figure 1. Running gear for a typical tracked vehicle.

taken place well in the years of later part of the last century. These types of suspensions are under consideration for various futuristic tracked vehicles of the world for various applications, ranging from earth moving to combating.

The ultimate aim of providing a flexible support through the suspension is to negotiate the obstacles and ditches encountered during a cross-country run to provide a better ride comfort to the crew. During the operation, the suspension components encounter high resisting loads in dynamic conditions and are subjected to higher level of vibration. While carrying out the kinematic analysis on the system, the study becomes vital due to the complexity of the forces acting on the components that form the suspension system like connecting rod, piston, seals, cylinders, etc. The suspension for a tracked vehicle encounters a dynamic load of about 120 kN to 200 kN with a gas pressure of around 400 bar to 800 bar, which acts here as a springing medium. The operating temperatures and frequencies are of high degree, challenging the designers to give importance for the kinematics design optimisation. Figure 2 shows the cut-section view of a 3-D modelled suspension.

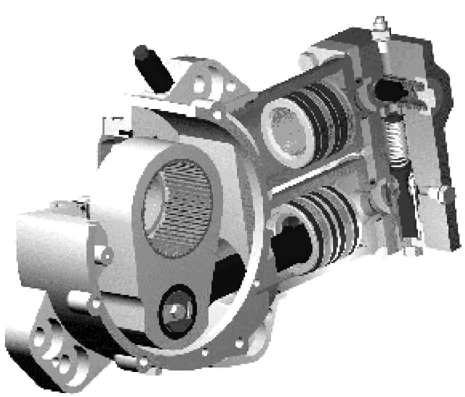

Figure 2. Cut-section model of a typical hydro gas suspension.

In the present study, the important design considerations to configure the kinematics with suitable linkage arrangement are considered wrt the loading pattern and operational requirements. The methodology adopted is quite adoptable for any similar kind of application, either for tracked vehicle or for other mechanical systems.

2. HYDRO GAS SUSPENSION

In hydro gas suspension system, instead of material torsional spring in a conventional mechanical torsion bar, gas is used to act as a spring medium. Hydraulic oil is filled to provide hydraulic damping to dissipate the energy, thus diminishing the successive amplitudes. A typical hydro gas suspension system which consists of a stationary casing, a rotating crank, a connecting rod and sliding pistons inside the cylinders are depicted in Fig. 3. The suspension is assembled with an axle arm and a hub is mounted over the arm through a stub axle. On this hub, road wheels are fastened to transmit the road input to the suspension. The casing, crank, piston rod and the cylinder form a four-bar single slider crank mechanism[2] as shown in Fig. 4, to convert the rotary movement of the axle arm to linear resulting piston displacement to compress the gas medium.

When the tracked vehicle is fielded into operation, the track drags up the vehicle utilising the engine power. Due to this, the suspension road wheels rotate, and when the vehicle negotiates an obstacle, the road wheel along with the axle arm swings up. The kinematic arrangement results in the displacement of the filled in hydraulic oil. Ultimately, this fluid compresses the accumulator piston against the entrapped nitrogen gas, thus raising the pressure of the gas to offer sufficient load. A damper is an

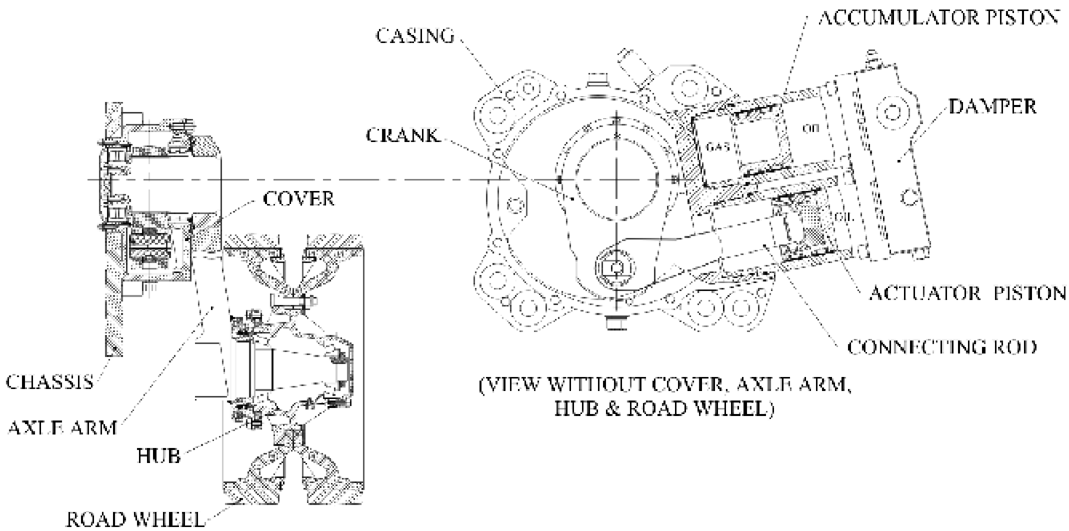

Figure 3. Internal arrangement of a typical hydro gas suspension system.

important subsystem of a suspension interposed in between the actuator and the accumulator to dampen out the vibration by dissipating the energy so as to eliminate the chances of resonance in the system.

In this suspension, gas is used as a spring medium and hydraulic oil is utilised for force transmitting and dampening out the oscillations. The advantage of employing the hydro gas suspension in a tracked vehicle is not only to isolate the primary vibrations induced into the suspension system but also to offer a better ride comfort through nonlinear springing of the gas. To take the maximum advantage of the suspension, it is expected that the system should have less spring rate, and at the same time, it should be able to vary according to the load. This is because at lower amplitudes, the behaviour of the spring should be very soft, whereas after two-third of the wheel travel, the spring should offer more resistance with high spring rate. This is because the suspension should have a minimum transmissibility ratio, but at the same time, it should be able to negotiate larger obstacles and gradients, and to counter the gun recoiling forces with high spring rate.

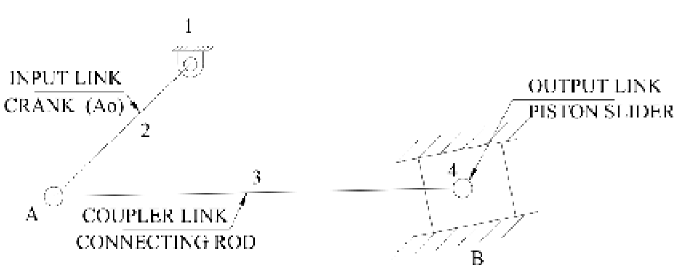

Figure 4. Four-bar mechanism of hydro gas suspension.

3. UNIQUENESS OF HYDRO GAS SUSPENSION IN KINEMATICS CONTEXT

Though the kinematic arrangement of hydro gas suspension represents a four-bar slider crank mechanism similar to that of reciprocating pump or an engine, the significance of loads, specifically the side thrust, assumes prominence. There is a major difference between other applications and the application of hydro gas suspension in providing near-zero leakage arrangement for the operating fluids. Unlike other systems, here the seals are made out of teflon material to handle high operating pressure, temperature and sliding velocity. In case, the side thrust due to the crank mechanism exceeds, the seal material will get deformed, resulting in the passage of fluid from one chamber to the other. This will lead to the failure of the hydro gas suspension system. Hence, there is a need to minimise the side thrust as far as possible by suitably designing the linkage mechanism and reorienting the axis of the cylinder, where the fluid displacement takes place.

4. LOADING ON SUSPENSION KINEMATIC LINKAGES AND DESIGN CONSIDERATIONS

The kinematics of hydro gas suspension is highly dependent on the loading pattern. Various factors affecting the kinematics configuration are:

Dynamic load acting vertically due to road input and acceleration/ braking

- Dynamic load acting axially due to turning manoeuvre and slope of the obstacle contour
- Static load due to vehicle weight
- Speed of the vehicle at which it operates as an input in terms of time
- Terrain conditions due to obstacles and ditches, resulting in amplitude input to the kinematic chain.

By taking the above detailed loads and amplitudes, the suspension has to perform its intended functions as follows:

- To convert the road wheel & axle arm rotation into linear displacement of piston
- Force transfer to the compensating energy absorbing medium
- To have minimum side thrust in turn to reduce seal side loads
- To effect larger fluid compression for softer nonlinear spring rate
- Discharge the fluid to effect required damping
- Absorbs shocks and vibrations due to lateral movement of the linkages
- To provide intended vertical wheel displacement
- To have optimum linkage load transfer factor to effect optimum level of damping.

The dynamic load acting on the suspension is complex owing to the input forces occurring due to vehicle acceleration, braking, turning manoeuvres, gun recoiling, ballistic impact, etc. The mass and speed of the vehicle offer higher inertial forces, and the kinetic energy varies directly with the mass and with square of the speed[1]. Due to this, heavy forces are set up in the suspension system, which are to be encountered with appropriate design of the kinematic linkages. Also, the centre of gravity due to variation in mass distribution of the vehicle plays a vital role in offering the loads to the suspensions. Figure 5 shows a schematic arrangement of the linkage mechanism for a typical hydro gas suspension.

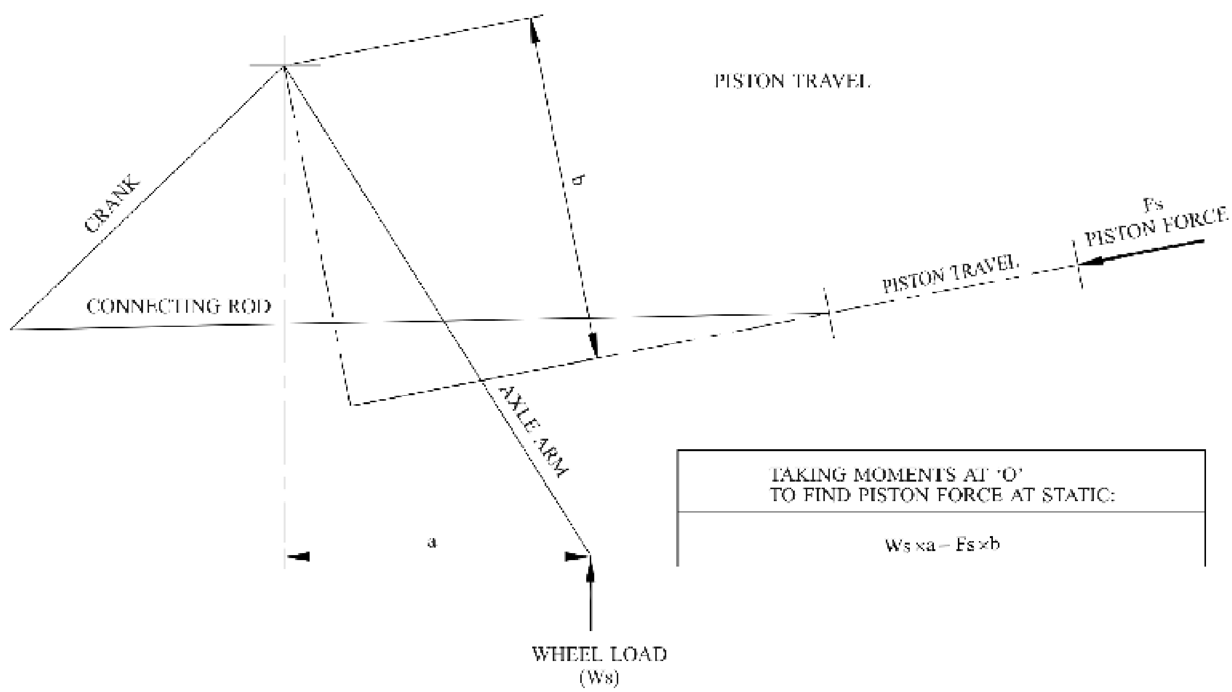

Figure 5. Kinematic arrangement of a typical hydro gas suspension.

As the loading pattern and requirements pose certain challenges, the design configurations and optimisation are carried out with the following considerations:

- The linkage members should be sufficient to transfer the forces
- Kinematic mechanism on trailing arm wheel lift configuration
- Maximum rebound and bump travel for the wheel
- Minimum unsprung mass for the dynamic components
- Occupying less volume of space.

The design of the suspension linkage starts with the known form of input and the ultimate output requirements. Table 1 depicts the input and output parameters to be considered for a basic kinematic chain meant for a hydro gas suspension:

Table 1. Input and output parameters for a basic kinematic chain meant for hydro gas suspension

Input (wheel)	Output (accumulator thro actuator)
Rotation angle (θ)	
Displacement (*y*)	Fluid displacement (*Qt*)
	Gas spring compression (ΔV)
Time (*t*)	
Frequency (*f*)	Fluid displacement rate (*Q*)
	Acceleration level (*g*)
Static wheel load (*Ws*)	Resisting force at piston (*Fs*)
Spring energy absorbed (*Fd*)	Wheel reaction load at dynamic (*Wd*)

5. EVOLVING KINEMATIC ARRANGEMENT FOR HYDRO GAS SUSPENSION

Tracked vehicles hydro gas suspensions are of different types, viz., rotating crank, fixed crank, arm and strut-in-tandem, etc. The operating principle is almost similar in nature other than the constructional arrangement. In this study the discussion is on the rotating crank, ie, stationary cylinder/casing hydro gas suspension.

The suspension is expected to offer more wheel lift to overcome high obstacle and also to take care of heavy dynamic loads during running condition. A compromise has to be made between the number of suspension stations to be fitted and the overall length of the vehicle. This results in dictating the volume availability for individual suspension, and the swing of the wheel arm is thereby limited. Knowing the space availability, the configuration studies are to be carred out, and accordingly, the kinematic analysis has to be carefully carried out to optimise the design parameters of the individual components. Based on the above, the following points have to be given paramount importance for effective design of the suspension kinematics:

- More wheel lift is desirable to have less vibration transmissibility, in turn, reduced acceleration level to the crew. The vertical wheel lift requirement dictates suspension axle arm rotation. But the axle arm rotation is restricted by the gap between the two suspension stations. On the other side, the gap between the stations is limited because the number of suspension stations, which is already fixed by the hull length and the requirement of lower mean maximum pressure (MMP). Also, the wheel lift is limited by the clearance between the track top portion and the suspensions bump position.
- The static forces set up in the suspension system will be known at static condition from the vehicle mass. The general design practise is to consider about 4 to 5 times the static load in dynamic conditions for a cross-country mobile tracked vehicle, which is quite sufficient to take care of maximum forces arising out of acceleration, braking and steering.
- Having known the maximum wheel lift and the forces which will be experienced by the system, the kinematic mechanism should be formulated within the available volume for suspension system.

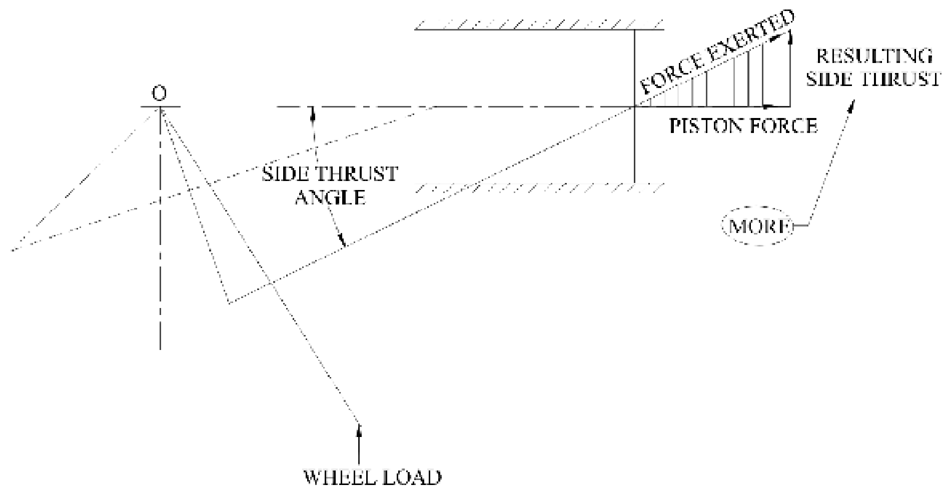

Figure 6. Effect of side thrust before shifting cylinder axis.

At this juncture, stress should be given on the strength of the components, which should be able to handle heavier dynamic loads. Other aspect to be borne in mind while formulating the kinematics is that the unsprung mass has to be kept at the minimum.

6. MINIMISATION OF SIDE THRUST

When the axle arm is made to rotate by the obstacle encountered, the crank rotates to an angle, which in turn forces the piston rod against the actuator piston. This pushes the actuator piston in the actuator. Due to the actuator arrangement, the piston rod axis and the cylinder axis intersects at an angle, which in turn offers a side thrust on the cylinder walls by the piston due to the force component. But the kinematics makes considerably a larger angle inducing more side thrust, which is highly undesirable for the teflon-based high-pressure seals as shown in Fig. 6. But the pivot axis of the crank and the axis of cylinder can be re-positioned to a distance equal to the vertical shift to obtain a reasonably minimal side thrust. It is shown in Fig.7 and the reduction of side thrust can be well-observed.

However, there exists a scope to reduce further the side thrust by reorienting the cylinder axis so that it makes a little intersecting angle with the piston rod axis. The optimum position of the piston rod axis is arrived at by taking the maximum operating frequency zone of the wheel. The reconfiguring of the cylinder position drastically reduces the side thrust which would otherwise lead to malfunctioning of the piston seals. This exercise definitely enhances the overall suspension performances, as the seal is one of the major contributors for the system life. Figure 8 pictorially depicts the effect of reorientation.

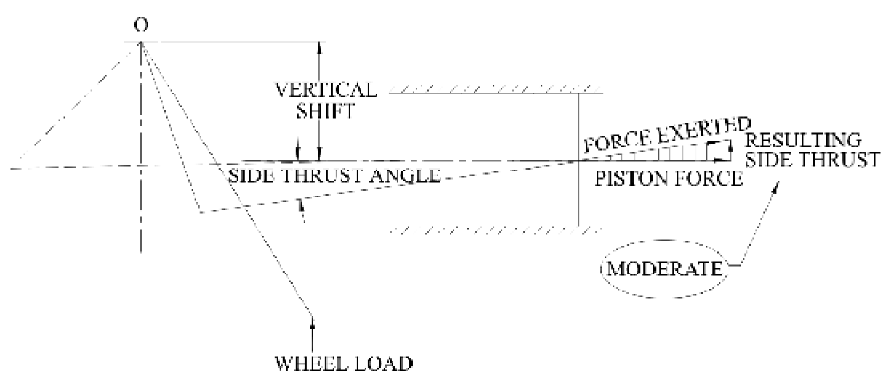

Figure 7. Effect of side thrust with cylinder offset but without reorientation.

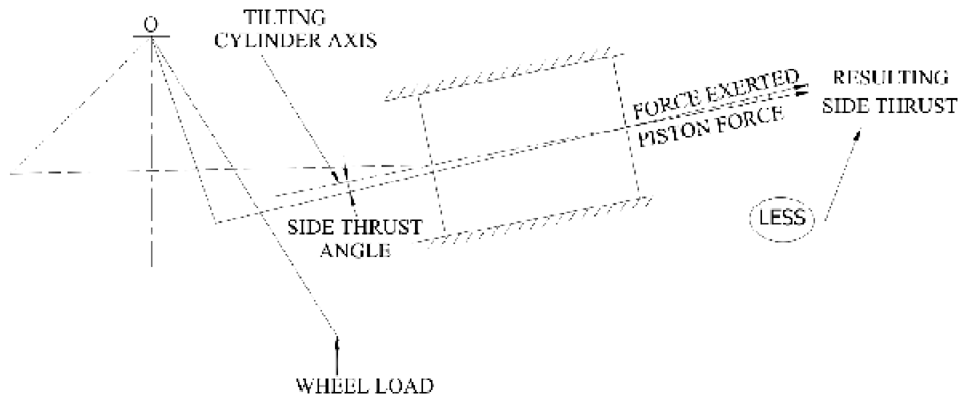

Figure 8. Effect of side thrust with cylinder offset and reorientation of cylinder.

7. OPTIMISING LINKAGE LOAD TRANSFER FACTOR

It is obvious that there exists a definite relationship between the force induced and the force transferred due to the fixed kinematic linkage movements. This is known as load transfer factor (LTF) and is the ratio between the wheel lift and the piston displacement.

$$\mathrm{LTF} = \frac{\text{Piston displacement}}{\text{Wheel travel}} < 1 \qquad \text{[Kinematic ratio]}$$

The crank length, piston rod length, and distance between the pivot point and the cylinder axis are the factors determining the load transfer factor. As this factor is primarily governing the gas compression volume and fluid discharge rate, it should be fixed at an optimum level for achieving better ride performance.

In general, the damper is an externally mounted subsystem of the suspension, but in hydro gas suspension, it becomes an in-built subsystem by introducing it between the actuator and the accumulator. For damper, the fluid discharge rate *(Q)* forms a major input to achieve the dampening of the successive amplitudes. The discharge rate is the function of time and piston displacement. The damping force *(Fo)* is directly proportional to the velocity and the fluid discharge rate should be kept at minimum. But for a better spring rate from a nonlinear gas spring, the volume of gas being compressed should be more. So the parameters like damper orifice area, piston force in dynamic condition (*Fd*) and change in gas volume (ΔV) are fine-tuned in tandem to arrive at an optimum load transfer factor. The sliding velocity of the piston seal is determined by LTF, which should be optimised, as the seal sliding velocity limits the life and sealing efficiency. In Fig. 9, the load transfer factor has been pictorially depicted.

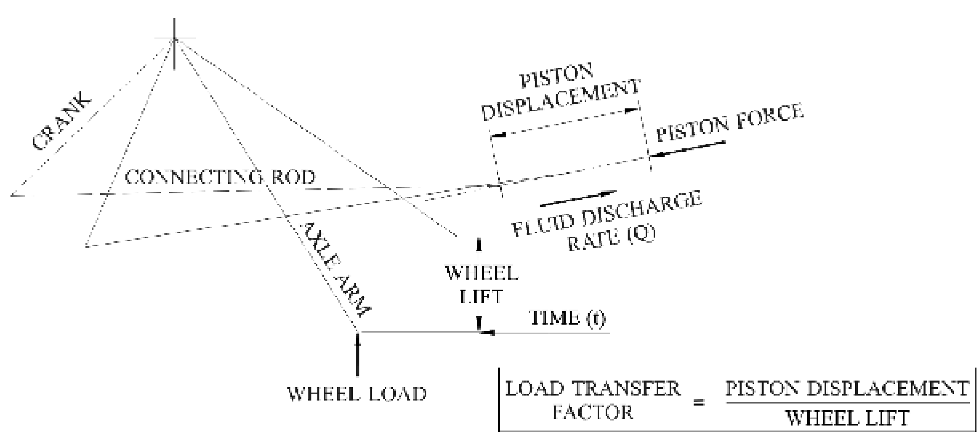

Figure 9. Load transfer factor for a given wheel lift.

As the damping force and the piston force in dynamic conditions are a function of load transfer factor, these are explained with the following governing equations:

7.1 Damping Force in Dynamic Condition

$$Fo = \Delta P * Ap$$

$$Q = Cd * Ao * \sqrt{(2g * \Delta P / \rho)}$$

where

Fo	Damping force
ΔP	Differential pressure
Ap	Area of piston
Q	Discharge rate
Cd	Orifice coefficient. of discharge
Ao	Area of orifice
g	Acceleration due to gravity
ρ	Specific gravity of the fluid medium

Therefore

ΔP is $f(Q, Cd, Ao, \rho)$ and Q is $f(t, Ap, l, \text{LTF}, y)$

where

l	Piston travel (difference of gas length between dynamic and static conditions)
t	Time
LTF	Load transfer factor
y	Wheel lift

7.2 Piston Force in Dynamic Condition

$$Fd = Pd * Ap$$

$$Pd = Ps(Vs/Vd)^n$$

$$Vs = Ap * ls$$

$$Vd = Ap * ld$$

$$ld = ls - l$$

$$l = LTF * y$$

where

Fd	Piston force in dynamic condition (for a given wheel lift)
Pd	Pressure at dynamic condition
Ps	Pressure at static condition (Fig. 5)
Vs	Volume of gas at static condition
Vd	Volume of gas at dynamic condition
n	Polytropic index
ls	Gas length at static condition
ld	Gas length at dynamic condition

Therefore

Fd is $f(Pd)$, *ie*, $f(Ap, \text{l}, \text{LTF}, t, y)$

Thus, load transfer factor, the kinematic linkage ratio plays a vital role in the piston and damping forces. Therefore, optimum load transfer factor is achieved by the parameters, viz., axle, arm length, crank length, connecting rod length, distance between crank fulcrum & cylinder position, angle between vertical axis, crank and axle arm. Also, the static-to-rebound travel and static-to-bump travel of the wheel through axle arm are the primary considerations, which determine the piston positions governing the maximum, piston force due to the limitation of the seals.

It may be noted that the total piston travel is a predominant parameter that could alter the gas compression, which means, lesser the travel, harder the stiffness, and more the travel, softer the stiffness. Hence, the kinematic optimisation is very essential in the context of slider crank mechanism.

8. CASE STUDY

To evaluate the effects of optimisation, a typical case of a tracked vehicle suspension has been taken and the configuration has been modified accordingly. This was evaluated both theoretically and in laboratory testing. This system was encountering

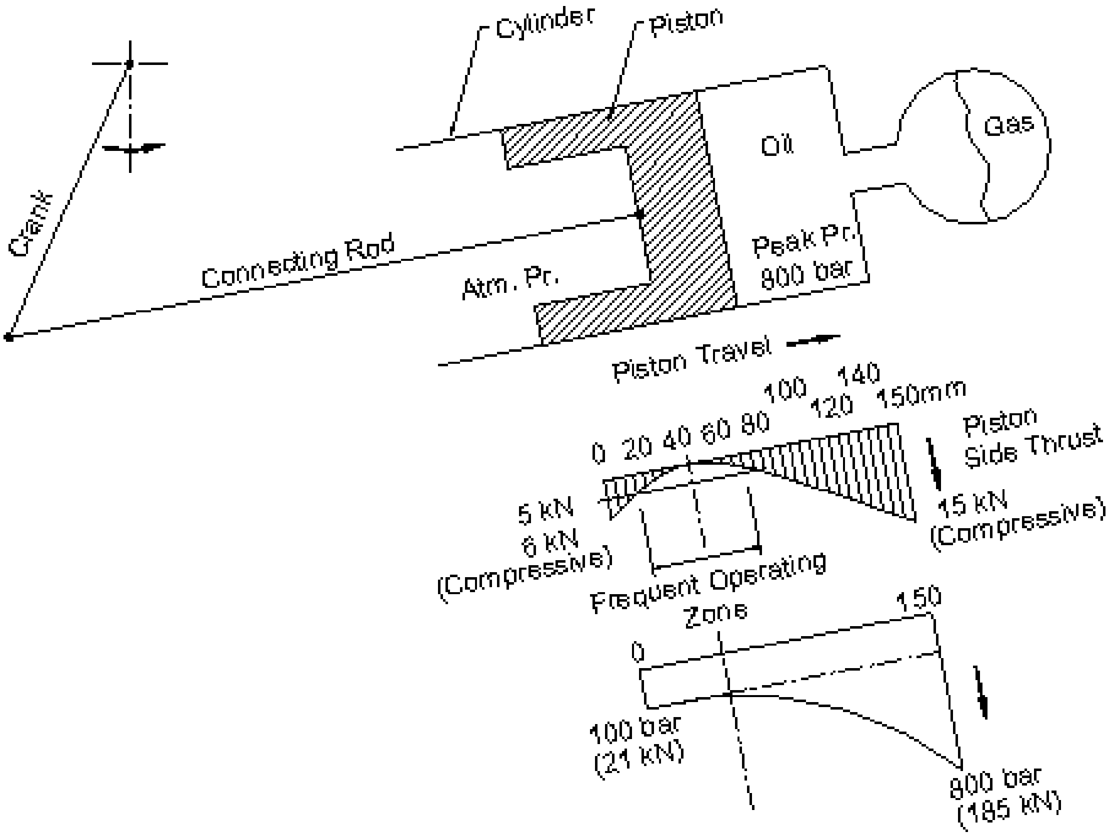

Figure 10. Suspension with optimised kinematics to offering minimum s[redacted]t.

a peak pressure of 800 bar during bump condition as the road wheel travels around 500 mm with 150 mm of piston travel with a load transfer factor of 0.3. The terrain conditions of a particular operating zone for the cross-country ride need mostly the piston to operate between 20 mm to 80 mm. The axis of the cylinder has been shifted by 120 mm vertically and reoriented by 10 ° to keep a minimum side thrust in the frequent operating zone. These kinematic changes resulted in a maximum side thrust of 15 kN, but in the operating zone, it was about 0 to 5 kN either direction, which is well within the acceptable limits. This has been depicted in Fig.10.

The suspension is fitted with hard chrome-plated cylinders inserted with composite PTFE-based seals mounted over the pistons. The seals and cylinders are the major elements of the suspension system dictating the life due to the function of sliding velocity, seal friction, system peak pressure, cylinder surface finish, and layer hardness and side thrust from the piston due to force transmitting structural members. The reduction of the side thrust ultimately reduces stress on the seal and the cylinder wall, resulting in longer fatigue life. The suspension system has been subjected for a range of vertical wheel frequencies from 0.1 Hz to 1.0 Hz with different amplitudes from 400 mm to 100 mm. The performance was found to be the multi-folded increase on account of the lesser side thrust.

9. CONCLUSION

In this study, an attempt has been made to highlight the significance of optimising the kinematics, particularly for a hydro gas suspension meant for a tracked vehicle. The uniqueness of the hydro gas suspension system has been dealt to understand the importance of carrying out the optimisation on the kinematics. A stress has been given to explain the difference of this particular application from the conventional practises. The methodology to evolve a kinematic arrangement for the referred application has been detailed with the loading pattern and some critical design considerations essential for the optimisation are highlighted. As the major threat for the suspension design is the sealing of high-pressure gas, the necessity and methodology for minimising the side thrust have been dealt in detail. Optimisation methods, viz., reorienting the cylinder axis, determination of load transfer factor, etc, are explained in detail.

Though this study essentially deals on the kinematic arrangement for a tracked vehicle suspension application, the methodologies explained in optimising various kinematic factors will be helpful while designing the similar mechanisms for other applications of this nature.

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