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An Active Suspension for a Formula One Grand Prix Racing Car

During 1982, Formula 1 racing cars generated very high downforces by the use of "ground effect" aerodynamics. Such cars required very stiff suspensions to maintain a reasonably constant ride height with the result that the slightest bump unsettled the chassis and reduced cornering speeds. A semi-active suspension would have been capable of withstanding the variations in downforce while remaining "soft" to rapid road inputs. This paper proposes such a system and decribes an analysis of its dynamic responses. It demonstates that it is able to maintain a sensibly constant ride height and attitude during cornering, braking, and acceleration, while minimizing the chassis response to individual bumps.

Introduction

During the 1982 Grand Prix season the performance of Formula One racing cars reached a peak due to the use of "ground effect" aerodynamics. The use of the air passing under the car to generate a low pressure area caused the car to be "sucked" onto the ground thereby increasing the grip of the tyres. The total downforce generated by the car could be as high as 25 kN at a velocity of 75 ms^{-1}, giving rise to cornering accelerations approaching 4g. Under these circumstances the aerodynamic properties of the car dominated the handling and road holding characteristics on all but the very slowest corners. Up until the 1983 season, when revised regulations effectively eliminated ground effects, the design of the chassis, suspension, engine, and transmission were all compromised to maximize the downforce [1].

Since the Lotus team introduced ground effect aerodynamics in 1978 and 1979 [2] the suspension had, in addition to its conventional functions of providng a satisfactory ride while transmitting control forces between the road and the car, been required to maintain a relatively constant ride height so that:

(i) the skirts that sealed the low pressure area under the car remained in contact with the road (Fig. 1),
(ii) the aerodynamic properties of the car remained reasonably constant, and
(iii) the suspension charactertistics remained acceptable throughout the speed range.

As the downforce acting on the car during cornering could vary from 1.5 kN at 18 ms^{-1} (at Monaco) to 32 kN at 85 ms^{-1} (at Ricard), conventional suspension using coil springs and hydraulic dampers had become very stiff, and wheel movements were minimized in an attempt to keep the skirts on the ground. Wheel rates of 450 kN m^{-1} and total movements of only 2.5 cm were not unknown. The result of such specialized suspensions was that although the cars could accept the wide variations in downforce experienced on many circuits, their response to bumps was extremely harsh. This made the cars very difficult and uncomfortable to drive.

As these very stiff suspension evolved it became apparent that an active or semi-active suspensions capable of responding to the changes in load was needed. It would have to maintain the desired ground clearance as the aerodynamic downforce increased, and at the same time improve the ride thereby reducing the shock loadings imposed on the suspension and chassis. This in turn would reduce driver fatigue.

Comparison of Suspension Systems

Figure 12 illustrates the two suspension systems generally used on Grand Prix cars in 1982. The "Lotus" system was originally introduced to move the springs out of the air stream. As the aerodynamic downforce increased the top rocker had to become larger in section to resist bending. To counteract this, many teams copied the "Brabham/Williams" system which takes the bending out of the suspension arms thus reducing uncontrolled deflections and weight at the expense of a slight increase in drag. The Brabham system can also be arranged to give a rising rate characteristic.

Semi-active suspensions, in which the system can modify itself in response to ground or chassis movement, are not new.

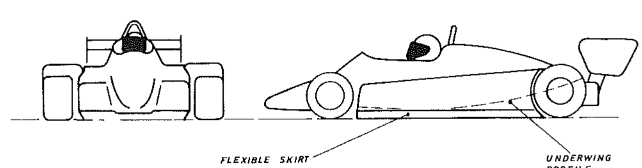

Fig. 1 Layout of a typical "ground effect" F.1 racing car

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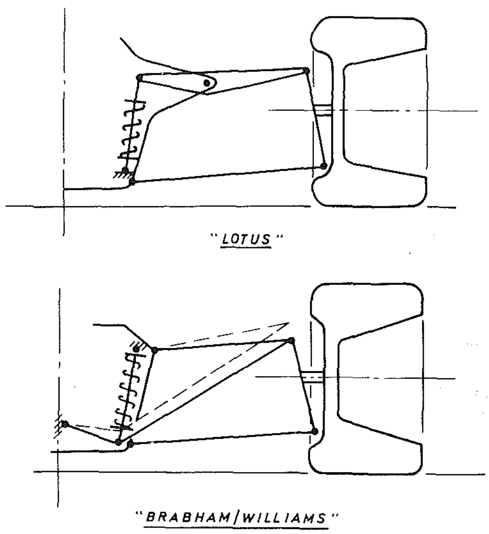

Fig. 2 Typical F.1. suspensions using co-axial coil springs and dampers

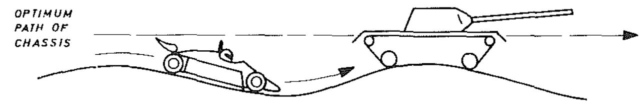

Fig. 3 Difference between active suspension requirements for a F.1. car and other applications

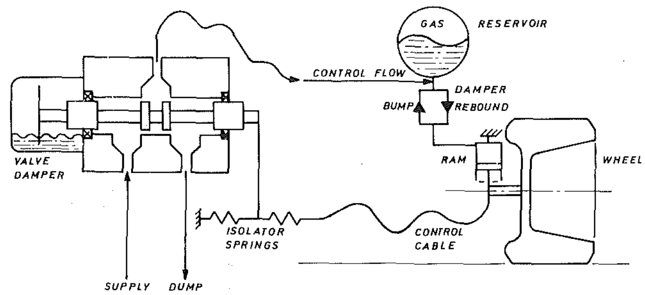

Fig. 4 Schematic diagram of semi-active suspension

Perhaps the best known system is that originated by the Automotive Products Group in the UK [3, 4]. This has been applied to vehicles ranging from small private cars to large military carriers. Unfortunately the system is relatively heavy and bulky for Formula One applications, and since it interconnects the front and rear wheels, it was considered to be too complex to allow easy chassis tuning. Furthermore there is a fundamental difference between the requirements of a conventional vehicle and a ground effect racing car. When a normal vehicle encounters undulations of any type the suspension is required to even out the bumps and keep the path of the chassis as close as possible to the horizontal. Much of the complexity of an active suspension is due to this requirement. However, in the case of a ground effect racing car the chassis must follow long wavelength undulations as closely as possible so that the skirts remain on the ground and the low pressure is maintained under the car (Fig. 3). At the same time the suspension must still absorb short duration inputs.

The regimes required of a racing car suspension can therefore be defined as follows:

(i) stiff, to ensure that loads imposed progressively due to roll, pitch, long wavelength bumps and changes in downforce do not change the attitude of the car,

and

(ii) soft, so that rapid suspension inputs due to short wavelength bumps are easily absorbed by the wheel motion and do not unsettle the chassis.

Such a suspension can be achieved by using a self levelling hydraulic unit at each wheel. The system can be adjusted to filter bumps below a certain wavelength yet react to long wavelength inputs to maintain contact between the skirts and the road.

The Semi-Active Suspension

The system specifically designed for a Formula One car is shown in Fig 4. As the wheel rises slowly its motion will be opposed by the increasing gas pressure in the reservoir. At the same time the wheel will cause the control valve to open, allowing high pressure fluid into the reservoir thus further increasing the gas pressure and tending to restore the wheel to its original position. If the wheel falls the control valve will allow fluid to leave the reservoir, reducing the gas pressure and letting the wheel return. Such a system will attempt to maintain the wheel at a constant position relative to the chassis.

Should the wheel be subjected to a very rapid input, the valve damper will tend to prevent shuttle movement and no control flow will occur. The response of the system will then depend upon the characteristic of the "gas spring" [5, 6].

The transition from one regime to the other will be progressive. As the period of the input reduces the response of the control valve will diminish.

The magnitude of the control flow is governed by the valve gain, and by the ratio of the stiffness of the two isolator springs. The time response of the valve is governed by the relationship between the degree of valve damping and the total stiffness of the two isolator springs.

Nomenclature

A = ram piston area
a = shuttle damper piston area
A_d = damper orifice area
A_{sh} = shuttle damper orifice area
C_d = damper orifice discharge coefficient
C_{dsh} = shuttle damper discharge coefficient
C_v = control valve discharge coefficient
F_x = externally applied force
h = area of control valve opening for a unit shuttle displacement
k_1, k_2 = isolator spring rates
m = mass (chassis)
n = expansion index
P_g = gas pressure
P_s = strut pressure
P_0 = reservoir precharge pressure
P_z = control valve supply pressure
q = suspension damper flow volume
q_{sh} = shuttle damper flow volume
r = valve gain (dependent upon control cable linkage ratio)
s = shuttle displacement
t = time
V_c = control flow volume
V_g = gas volume
V_0 = reservoir free gas volume
x = wheel vertical displacement
x_m = mass (chassis) vertical displacement
x_i = input (ground) vertical displacement
ρ = working fluid density

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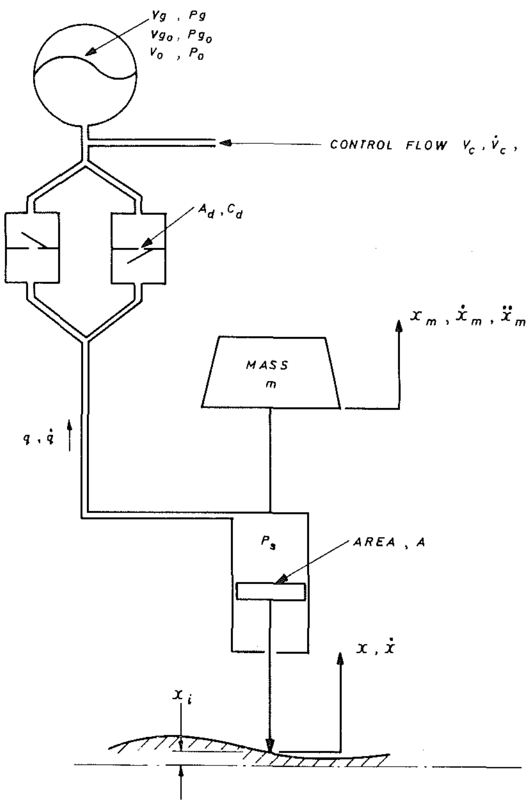

Fig. 5 Hydraulic strut model

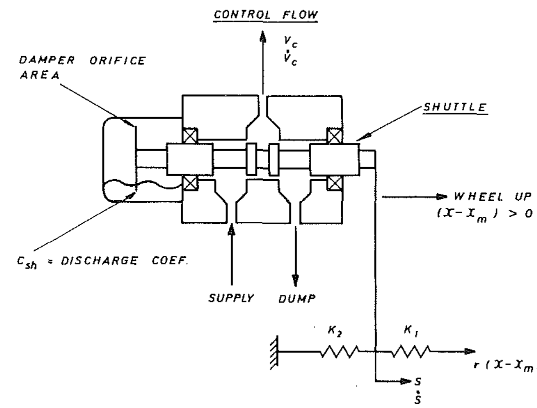

Fig. 6 Control valve model

The Analysis

The analysis considers the response of a single suspension unit to both chassis and road inputs. For simplicity the unit is considered to act directly on the wheel hub (i.e., no mechanical advantage) and the compliance of the tyre has been neglected for two reasons. Firstly, it is the objective of the analysis to consider the action of the active system alone and, secondly, the high effective spring rates and relatively low unsprung masses typical of the Grand Prix car result in wheel hop frequencies much higher than those of interest to this work. A further synthesis of the response of the complete chassis would include tyre effects.

The analysis effectively solves three symultaneous, nonlinear systems; the spring-mass system, the control system, and the control flow. In this case a time step solution has been used since this is an easily understood technique that operates well within the limitations of the available microcomputer.

The Hydraulic Strut. Consider the diagramatic system shown in Fig. 5. When $P_s > P_g$;

$$\dot{q} = C\sqrt{\Delta P} \tag{1}$$

where $C = A_d C_d \sqrt{\dfrac{2}{\rho}}$

$$\therefore \dot{q} = C\sqrt{P_s - P_g} \tag{2}$$

Now for a polytropic process, and if P_{g0} and V_{g0} are the pressure and volume of the reservoir at the beginning of the time step; then

$$P_g V_g{}^n = P_{g0} V_{g0}{}^n \tag{3}$$

and for a flow q, due to movement of the piston, and a control flow V_C,

$$P_g = P_{g0} \bullet \left[\frac{V_{g0}}{V_{g0} - q - V_c} \right]^n \tag{4}$$

If we express equation (2) as

$$P_s = P_g + \left(\frac{\dot{q}}{C} \right)^2 \tag{5}$$

and note that

$$\dot{q} = (\dot{x} - \dot{x}_m) A$$

then

$$P_s = P_g + \left(\frac{(\dot{x} - \dot{x}_m) A}{C} \right)^2 \tag{5}$$

On substituting (4) into (6)

$$P_s = \left[\frac{(\dot{x} - \dot{x}_m) A}{C} \right]^2 + P_{g0} \left[\frac{V_{g0}}{V_{g0} - q - V_c} \right]^n \tag{7}$$

and for a time step δt,

$$q = (x - x_m) A = (\dot{x} - \dot{x}_m) A \; \delta t \tag{8}$$

and

$$P_s = \left[\frac{(\dot{x} - \dot{x}_m) A}{C} \right]^2 + P_{g0} \bullet \left[\frac{V_{g0}}{V_{g0} - (\dot{x} - \dot{x}_m) A \bullet \delta t - V_c} \right]^n \tag{9}$$

A similar expression is obtained when $P_g > P_s$ and these can be used to give the vertical acceleration of the mass, i.e.:

$$\ddot{x}_m = \frac{P_s A + F_x}{m} \tag{10}$$

where F_x is an external (in this case aerodynamic and gravitational) force. Thus for a total time, t, the velocity and displacement can be found.

Control Valve Movement. Consider the value configuration shown in Fig. 6. If the shuttle has a negligible mass, then for equilibrium:

Damper force = Spring force

i.e. $$\Delta p \bullet a = [r(x - x_m) - s] k_1 - s k_2 \tag{11}$$

Now $$\dot{q}_{sh} = C_{dsh} \bullet A_{sh} \bullet \sqrt{\frac{2\Delta p}{\rho}}$$

also $$\dot{q}_{sh} = \dot{s} a$$

These can be rearranged to give

$$\dot{s} = \frac{C_{dsh} A_{sh}}{a} \sqrt{\frac{2}{\rho a} \{ [r(x - x_m) - s] \bullet k_1 - s k_2 \}} \tag{12}$$

At the end of a time step δt the valve position is;

$$s = s_0 + \dot{s} \bullet \delta t \tag{13}$$

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Table 1 Data table

Data used in analysis unless otherwise specified in the test

Ram piston diameter		.05 m
damper orifice diameter	bump	.004 m
	rebound	.002 m
Control valve opening for a unit displacement of the shuttle		.006 m
Isolator spring rate	k_1	150 N m^{-1}
	k_2	150 N m^{-1}
Exapnsion index		1.4
mass		150 kg
Reservoir volume		0.5×10^{-3} m^3
Precharge pressure		1 bar

All discharge coefficients are assumed to be 0.7.

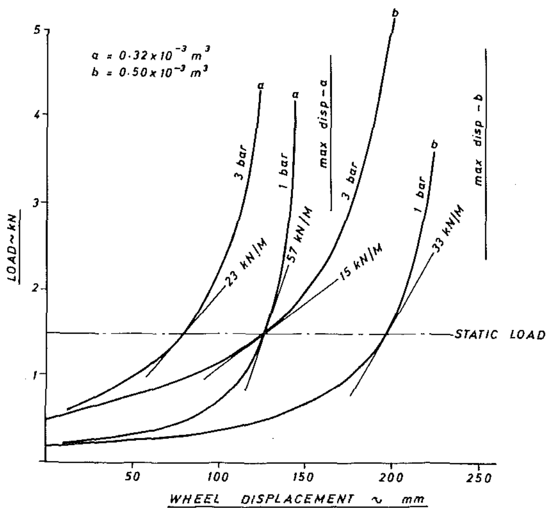

Fig. 7 Load-displacement characteristics for a 0.32×10^{-3} and 0.50×10^{-3} m^3 capacity reservoirs

Control Flow. During a time step δt,

$$\Delta V_c = \dot{V}_c \, \delta t \tag{14}$$

and $$\dot{V}_c = C_v sh \cdot \sqrt{2(P_z - P_g)/\rho} \qquad \text{for } s \geq 0 \tag{15}$$

$$\dot{V}_c = -C_v sh \cdot \sqrt{2P_g/\rho} \qquad \text{for } s \leq 0 \tag{16}$$

Power Consumption. With any active or semi-active suspension system it is important to know the power requirements. In this case, the work done for a time step δt is given by:

$$\text{Work done} = \text{Pressure} \times \text{Volume}$$

$$= P_z \cdot \Delta V_c \text{ when } s > 0 \tag{17}$$

and $$= 0 \quad \text{when } s \leq 0 \tag{18}$$

The instantaneous power when $s > 0$ is therefore

$$\frac{P_z \cdot \Delta V_c}{\delta t} \tag{19}$$

Equation (19) gives an instantaneous power requirement which can be used to give the mean power over a particular time period. In practice an hydraulic accumulator would be used to even out the pump power.

Application of the Model

The above equations represent the basic operating characteristics of the supsension system. The complete mathematical model, which was used to determine the vehicle responses, allowed for isothermal compression of the gas while the vehicle found its static ride height. The dynamic process was adiabatic, i.e., $n = 1.4$. A limited suspension movement was also included so that the case where the wheel might leave the ground was considered.

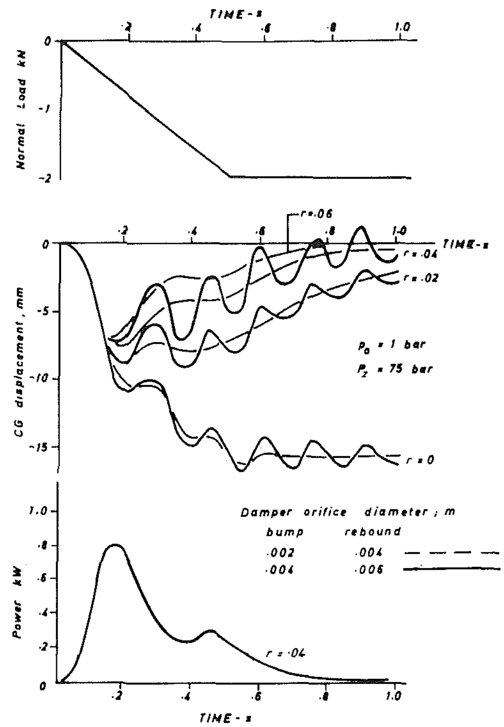

Fig. 8 Semi-active suspension response to an increasing downward load

The following inputs were used;

(i) a progressively increasing downward load or ramp input,

(ii) single sinusoidal bumps, and

(iii) continuous sinusoidal undulations.

The design parameaters are given in Table 1. These were chosen after consideration of suspension and damper rates is an existing F1 car. One important point concerns the choice of reservoir volume and precharge pressure.

Since the suspension relies on compressed gas as its springing medium, the reservior precharge pressure and volume governs the "springing" characteristics. Figure 7 shows a force displacement diagram for two precharge pressures (3 bar and 1 bar) and two reservoir volumes, (0.5×10^{-3} m^3 and 0.3×10^{-3} m^3). It can be seen that for a given static load, raising the precharge pressure or increasing the volume reduces the effective spring rate. This offers a powerful means for adjusting the basic suspension characteristics. It is perhaps a good reason for also choosing air springs as a springing medium on passive suspension systems.

Ramp Input. The object of the semi active suspension is that it should maintain an almost constant ride height under conditions of varying load. The case of the gradually increasing load corresponds to weight transfer onto a wheel during braking and cornering, or the increase in aerodynamic downforce as the car accelerates. Figure 8 show the response to a 2kN load progressively applied over a 0.5s period. The ride correction becomes faster as the valve gain, r, is increased although values greater than 0.06 do not give a significant improvement. The degree of damping is important; too little allows an oscillation to occur while too much slows down the response.

Single Bumps. The system response to a single sinusoidal bump of 0.02 m amplitude and 5m wavelength is shown in

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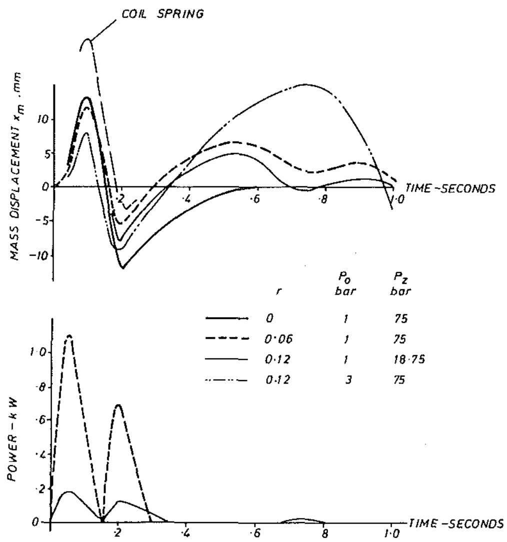

Fig. 9 Semi-active suspension response to a 0.02m amplitude bump of 5m wavelength at 30MS $^{-1}$

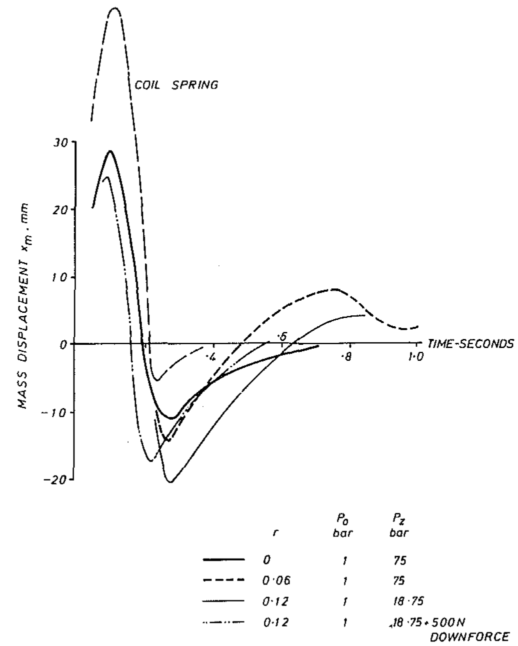

Fig. 10 Semi-active suspension response to a 0.02m amplitude bump of 1m wavelength at 30MS $^{-1}$

Fig. 9. The car velocity is 30 m.s. $^{-1}$ and there is no downforce. With a precharge pressure of 1 bar and a supply pressure of 75 bar an increase in valve gain reduces the suspension compression following the bump but causes a severe overshoot. As the gain increases from 0 to 0.06 the compression falls from 13 mm to 5 mm but the subsequent overshoot rises from 0.75 mm to 7.0 mm.

Raising the precharge pressure to 3 bar and increasing the valve gain to 0.12 reduces the initial response to the bump but allows more compression due to the softer wheel rate. The overshoot has a tendency to become unstable with relatively high valve gains. This is a consequence of an imbalance between the bump and rebound flow conditions through the valve.

In an attempt to improve the balance the gain was doubled from 0.06 to 0.12 while the supply pressure was reduced by a factor of four, from 75 bar to 18.75 bar. The response is good, with an acceptable compression and a much improved recovery to the datum position. It is very difficult to eliminate the oscillation completely because the orifice plate dampers give a damping force proportional to the square of the fluid velocity whereas conventional automotive dampers give a force that is linearly proportional to the velocity. (This could of course be included.) It can be seen that the coil spring/damper suspension gives almost twice the vertical displacement of the active suspension over the initial bump. The coil spring stiffness chosen gave a similar transient deflection to that shown in Fig. 8 for a valve gain of 0.06. This was 280 kN/m, and is typical of the rates used in 1982 Grand Prix cars.

The response to a 1m wavelength bump again of 0.02 m amplitude and taken at 30 ms $^{-1}$, is illustrated in Fig. 10. In this case the bump is considerably shorter than the wheelbase of the car and the active suspension would not be expected to respond in any significant way. As the gain is increased from 0 to 0.06 there is little change in behavior showing that the system does indeed act as a soft, hydraulic suspension. A bump of this severity causes the wheel to become well clear of the ground. When the gain is increased to 0.12 and the supply pressure dropped to 18.75 bar this presents a diffculty. While the wheel is in the air the control valve lets fluid leave the reservoir in an attempt to raise the wheel back to the chassis. As the wheel touches the ground, the gas pressure in the reservoir is low and the suspension is unable to resist the momentum of the chassis which results in a large compression.

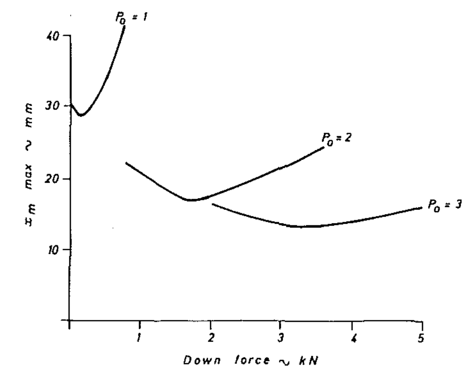

Fig. 11 Effect of aerodynamic downforce on suspension response to a 0.02m amplitude by 5m wavelength bump at 30MS $^{-1}$

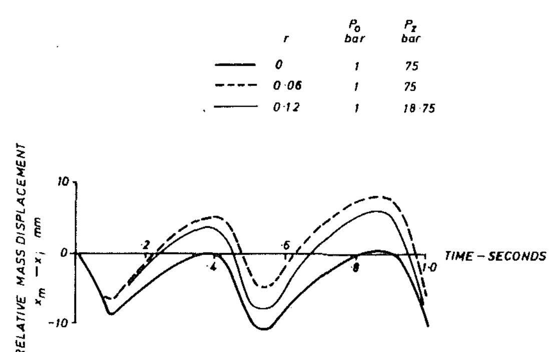

Fig. 12 Response to a 0.05m amplitude by 25m wavelength sinusoidal undulation at 50MS $^{-1}$

In the real case the car would be under the influence of considerable aerodynamic downforce. The addition of a downward force of 500N shows that both the deflection over the bump and the subsequent compression are brought back under control. However, at the low precharge pressures needed to achieve the required spring rates the gas volume in the reservoir is small. An increase in downforce requires that the system pump more fluid into the reservoir to maintain a

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constant ride height which further reduces the volume. As the wheel is pushed upwards by a bump the gas volume approaches zero and the effective wheel rate rises rapidly. This gives the chassis a high vertical force and consequently a large displacement. Figure 11 shows the effect of downforce on the peak vertical displacement as the car passes over a 5.0 m wave-length bump. As the downforce is increased the amplitude is initially reduced until the free gas volume beomes too small whereupon the mass is thrown in to the air. If the spring rate is reduced by increasing the precharge pressure, the gas volume is increased and more downforce can be applied. Little advantage is gained by increasing the pressure to more than 3 bar, (Fig. 11).

Undulations. Figure 12 shows the system's response to a continuous sinusoidal input of 0.05 m amplitude and frequency of 2Hz (25m wavelength at 50 ms^{-1}) without downforce. Under these conditions the suspension is required to minimize changes in ride height in order to keep the skirt in contact with the road surface. The supply pressure of 18.75 bar and valve gain of 0.12 which gave the optimum performance over the single bump again gives satisfactory results, successfully compensating for the undulations and behaving as a "stiff" conventional supsension. The effect is improved still further with the addition of downforce because of the rising rate of the gas spring.

Power Consumption. Figures 8 and 9 show that all times the power required to drive the suspension is remarkably low an would not be expected to affect the performance of the car significantly.

Valve Damping. Although the design and analysis of the suspension system included a damper on the control valve shuttle it has, so far, been found to be unnecessary. The required degree of filtering is achieved by the time taken for the control flow to have a positive effect on the system response. This is unlikely to be the case in other applications and the shuttle damper will again be required. Figure 9 shows that over a 5 m wavelength bump, where the chassis would be expected to follow the contour of the road, the system responds to changes in valve gain, the greater the gain the stiffer the suspension becomes. For the 1 m wavelength bump, where the suspension would be required to absorb the wheel motion, there is little change in response with gain and the suspension remains relatively soft, (Fig. 10), Clearly the active system is filtering out bumps shorter then the wheelbase of the car, which is the characteristic required.

Conclusions

The analysis of the semi-active suspension system described in this paper has demonstrated that such a system should successfully meet its design objectives in the following respects;

(i) the system will respond quickly to chassis and aerodynamic inputs to maintain a sensibly constant ride height.

(ii) the system allows rapid road inputs (i.e., bumps) to be absorbed by an apparently "soft" suspension therefore minimizing the forces tending to unsettle the chassis.

(iii) the system maintains a sensibly constant ride height over an undulating road surface.

Thus, a relatively simple semi-active suspension will accommodate large changes in aerodynamic and cornering forces without recourse to a conventional, very stiff system. In the real case the difficulties that have been encountered with the wheel leaving the ground over large amplitude bumps to a very short period will be reduced, firstly by the tyre, which will absorb some of the bump, and secondly by the aerodynamic downforce which will tend to push the wheel back onto the road.

It is acknowledged that a more versatile control function may be achieved using modern electronics, but the mechanical system offers a reliable suspension that can be easily developed in the context of a Grand Prix racing season. The analysis allows the optimization required for a particular application.

Although the 1983 regulations effectively eliminate "ground effects" on F1 cars this suspension system will be generally useful on any high downforce racing cars such as Formula 2, Group C (Le Mans sports cars) and Cam Am. A particulary interesting application would be "Indianapolis" type single seaters which not only generate very high downforces due to their peculiarly high cornering speeds (approaching 90 ms^{-1}) but also undergo additional suspension compression due to the steeply banked corners on the American oval circuits.

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