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Effect of road profile and curves generated by wheel on the performance of a shock absorber of a motorcycle

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Abstract

A front wheel suspension system of a motorcycle is assigned to a quarter car model and from which one shock absorber is considered for the performance study. The deviations from the cycloidal rotation of the wheel are proposed and the vertical motion of the wheel with respect to the sinusoidal road surface is modelled by a polynomial equation and compared with the experimental data. To give deterministic impulses due to road profile, the adjustable bumps are angularly mounted on a variable eccentric cam which gives input motion to a shock absorber mounted upright above it and acting as follower to the cam. For the heavier loads the rebound of shock absorber reduces and displacement smoothens. There occurs a combination of speed range, load and road disturbances within which the vertical acceleration is within the standard limit.

Keywords: Quarter car model, shock absorber, road profile, curves generation by wheel.

Introduction

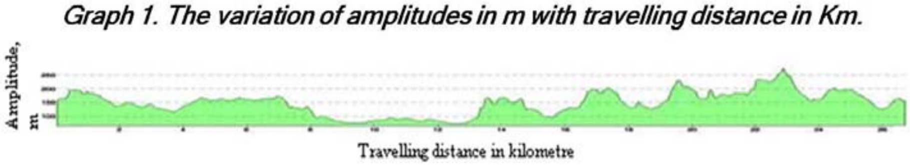
A two wheeler vehicle is probably the most challenging type of vehicle in terms of sensitivity to suspension tuning, since it has a comparatively short wheel base and high centre of gravity. Moreover, especially for the front suspension, the direct link of the handlebar to the front suspension fork makes the driver extremely sensitive to any little modification of the suspension characteristics. The two wheeler represents a half car model *i.e.* two coupled quarter car model (Imine *et al.,* 2003). The vehicle is subjected to variety of forcing conditions due to road profile. A passive suspension is subjected to various road conditions like- a single step road profile, brake and release maneuver, sinusoidal road profile with pitching, heaving and mixed mode excitations, broad band road profile etc at constant or variable speeds. A road bump gives a transient force (i.e. a non-periodic but deterministic excitation giving an impulse or shock) or a random excitation (unpredictable time and must be described in terms of probability & statistics). The deterministic impulses due to road may be experimented in laboratory and unpredictable disturbances and combinations of loads may be tried in field (Savaresi & Spelta, 2009).

Road profile & wheel travel

A road is considered as an infinite cam with the wavy profile of mixed harmonic sine waves and the wheel of the vehicle may be considered as a follower having vertical movement only (Graph 1). Out of number of superimposed sine curves, one simple wavy curve may be considered on which a quarter car model travels (Graph 2). A front wheel suspension system of a motorcycle is assigned to a quarter car model and from which one shock absorber is considered for the performance study as a decoupled assembly (Fig. 1). The quarter car model suspension considers the conventional inertia force due to sprung mass M_b with the displacement X_b and the shock absorber resistance comprising (1) the spring force K_s and the (2) resistance force due to dashpot C. For simplicity of experimentations the displacement of unsprung mass M_t, with displacement Xr and tyre stiffness Kt is not considered and only the effect of road profile on the shock absorber is considered as shown in Fig. 2. The base motion displacement of the shock absorber is $x = X \sin \omega_1 t$. The base motion of the shock absorber is the motion of the axle holding the wheel. Under the 200 kg static load the motorcycle wheel axle undergoes static deflection of 0.030 m. Graph 3 represents the displacement in static loading and before any dynamic condition. The static deflection takes some time to be stable.

Assuming the road profile approximated by a sine wave of amplitude = 0.0125 m and a wave length of 14 m. the natural frequency of the shock absorber, $\omega_v=\sqrt{(g/\text{static deflection})}$ =18.083 rad/s i.e. 2.878 cycles/s. The corresponding forcing frequency ω for a speed of 20 km/h = 2.513 rad/s. Assuming damping factor ζ = 0.35 , the amplitude due to road excitation = $X[\sqrt{\{1+(2\ \zeta\ \omega/\omega_n)^2\}} \div [\sqrt{\{1-(\omega/\omega_n)^2\}^2 + (2\ \zeta}$

Graph 1. The variation of amplitudes in m with travelling distance in Km.



Graph 2. The road is approximated as sinusoidal in cross section. Y axis = lift, m, X axis = distance on road m.

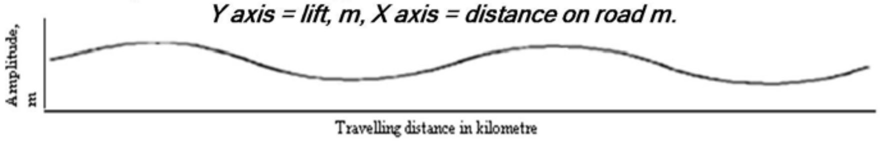


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Fig.1. Conventional quarter car model suspension

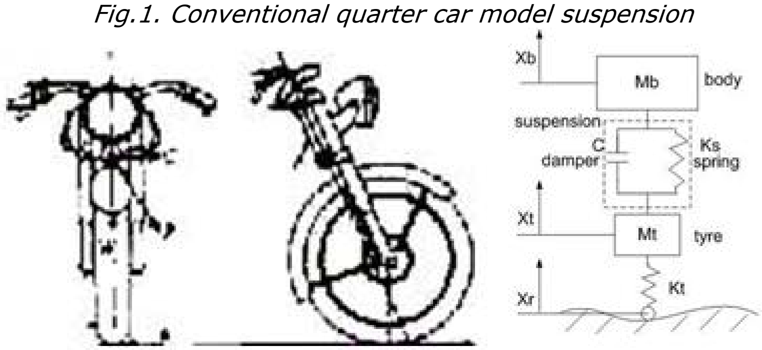
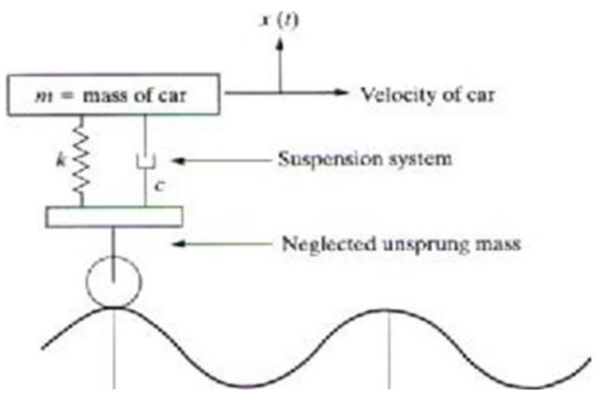
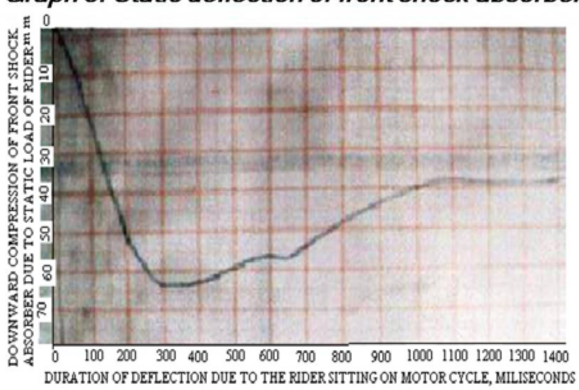


Fig. 2 . Simple suspension model of vehicle traveling at constant velocity on a approximated sinusoidal surface.



Graph 3. Static deflection of front shock absorber.



$\omega/\omega_n)^2$ }]] = 12.743 m (Inman, 2001), The amplitude changes from 12.5 mm to 12.743 mm. Assuming that the lift of the mass end of the shock absorber to be sinusoidal, the comparison between the two amplitudes may be made as per Table 1.

Except for the condition when damping in the system is more than 20% of the critical value, un-damped and damped natural frequencies will not differ by more than 2% and the un-damped natural frequency can be used without introducing any appreciable error. The above considerations are very much simplified and only harmonic smooth motion on the road is considered. The rotation profile needs further considerations. When the wheel rotates on the straight surface then any point on the circular tyre without any deformation generates a cycloid curve (Fenton, 1996) as per Fig. 3. A cycloid represents the changing linear speed of a point during each cycle of rotation. For the short time the point at ground level has zero linear velocity and at its highest position from the ground its forward velocity will be at maximum. The average forward velocity of the point is at mid point, i.e. at axle level which is also the vehicle's forward speed. The top of the tyre moves at twice the speed of the vehicle and in the same direction. The generating point will accelerate from zero to a maximum velocity for half a revolution and then decelerate to zero velocity again to complete the second half revolution. Since the point or the spot on the tyre has a mass and changes it's velocity it will be subjected to a varying acceleration force which acts in the direction tangent to the curve. Consequently the direction of the inertia pull caused by this heavy spot constantly changes as the wheel moves forward. Within very short duration heavy spot decelerates downward to ground, stops, changes its direction and accelerates upward. Hence at the end of each cycle and beginning of next cycle there will be a tendency to push down and lift up the tyre from the ground. Since the tyre material is assumed continuous the decelerating and opposing accelerating forces are balanced. But at higher speeds the magnitude of the accelerating force acting on any out of balance mass rises and there by produces the periodic bump and bounce or jerking response of the tyre. For any analysis of the suspension system it is assumed that the wheel assembly is statically and dynamically balanced.

The displacement of the wheel axle in vertical direction is as per the amplitude of the cycloid and is given by

$X_b = r\,[1-\text{Cos}\ \omega_2 t)$. r = wheel radius, mm

Therefore the vertical displacement of the base motion as input to the shock absorber is given by

$X_b = r\,[1-\text{Cos}\ \omega_2 t) + X\ \text{Sin}\ \omega_1 t$.

The tyre encounters with road bump or speed breaker and gets compressed and is flattened. Further due to load and variable air pressure in the tyre the circumference is flattened at the contact surface, one may consider a generating point to be inside a generating circle for the cycloid profile. The cycloid profile gets modified to a curtate cycloid (Fig. 4).

The tyre encounters with road pits and gaps between road's continuous bumps and that portion of tyre in the pit without surface contact gets stretched and is enlarged. One may consider a generating point to be outside a generating circle for the cycloid profile. The cycloid profile gets modified to a prolate cycloid (Fig. 5). The wheel rotates without slipping on a sinusoidal road profile and not on a straight line surface; therefore the curtate and prolate profiles are modified. For the generating circle moving on a convex surface the curves are modified to curtate epicycloid and prolate epicycloids (Fig. 6 & 7). For

Fig. 3. Cyclic movement of a point on wheel relative to road generating cycloid curve.

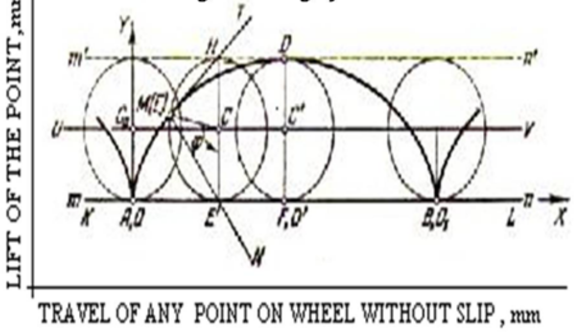


Fig. 4. Cyclic movement of a point on a flattened wheel relative to road generating curtate cycloid.

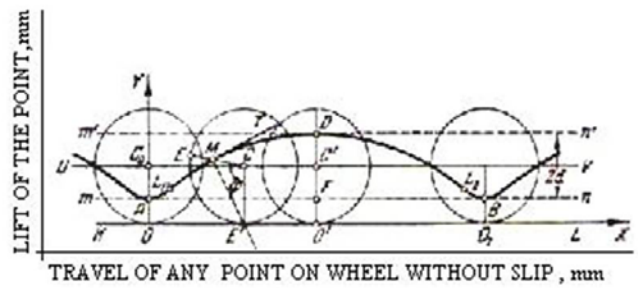


Fig. 5. Cyclic movement of a point on a stretched wheel relative to road generating prolate cycloid.

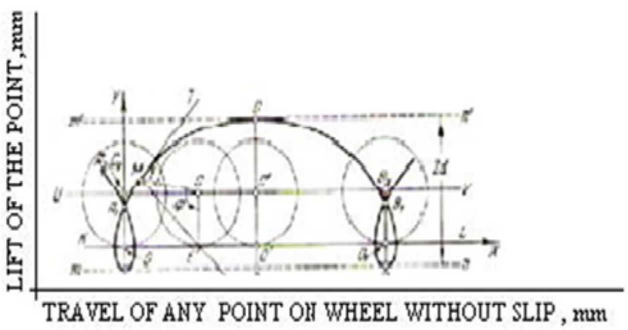


Fig. 6. Curtate epicycloids.

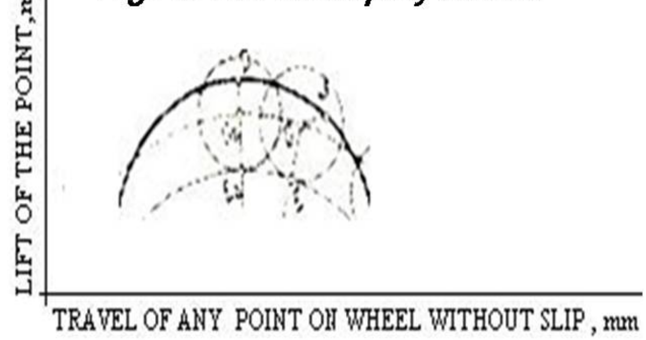


Fig. 7. Prolate epicycloids.

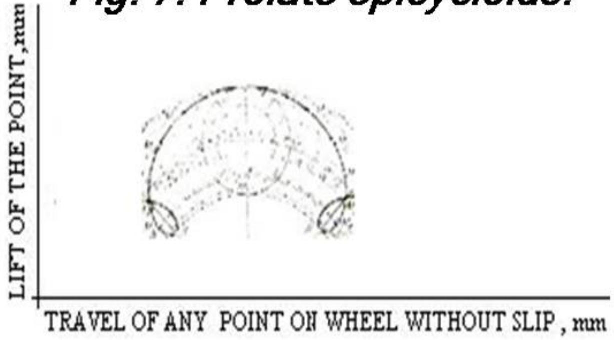


Fig. 8. Curtate hypocycloids.

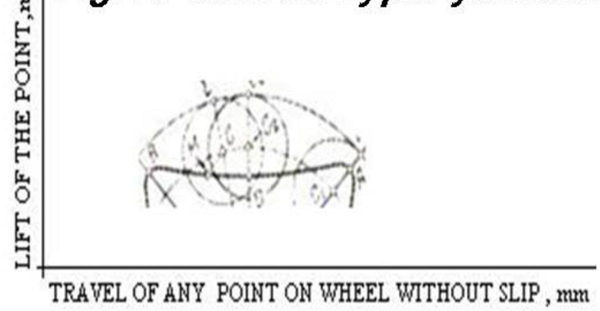


Fig. 9. Prolate hypocycloids.

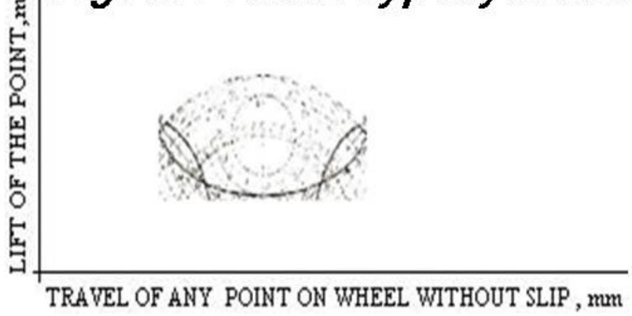


Fig. 10. Experimental set up.

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the generating circle moving on a concave surface the curves are modified to curtate hypocycloid and prolate hypocycloid (Fig. 8 & 9) (Vygodsky, 1978).

Table 1. Amplitudes of shock absorber as per the wheel rotation, θ^o.

θ^o	0	10	20	30	40	50	60	70	80	90
A: amplitude of wheel end of Shock absorber, mm.	0	2.17	4.27	6.25	8.03	9.57	10.82	11.74	12.31	12.5
B: amplitude of mass end of Shock absorber, mm	0	2.21	4.35	6.37	8.18	9.76	11.03	11.97	12.54	12.74

The response of the shock absorber: Wheel rotation on a wavy road surface thus gives the lift to the axle and therefore to the shock absorber in a combined and complex way. Due to the mixed travel mode of the wheel i.e. combinations of cycloids, curtates, prolates, curtate epicycliods, curtate hypocycloids, prolate epicycloids, prolate hypocycloids the solution of this equation may not be feasible as X_b becomes a complicated function. One method of rectifying this deficiency is to combine several options of these basic curves to obtain the displacement diagram. While doing this, the velocities and the accelerations should be matched at the junction points. Another method is to represent the basic shock absorber motion by polynomial curves (Ghosh, 1996), the displacement X_b is taken to be a polynomial in θ, the rotation of the wheel.

The response of the shock absorber:

$X_b = C_o + C_1\theta + C_2\theta^2 + C_3\theta^3 + \ldots + C_n\theta^n$ (1)

The number of terms to be taken is equal to the number of conditions to be satisfied.

$X_b = 0$, $dx/dt = 0$, $d^2x/dt^2 = 0$ at $\theta = 0$,

X_b = max. amplitude $= X$, $dx/dt = 0$, $d^2x/dt^2 = 0$ at $\theta = \theta_{rise}$,

X is obtained at the peak of the road bump for $\theta = \theta_{rise}$ of wheel rotation from start of bump till peak.

The terms up to C_5 is taken, which gives

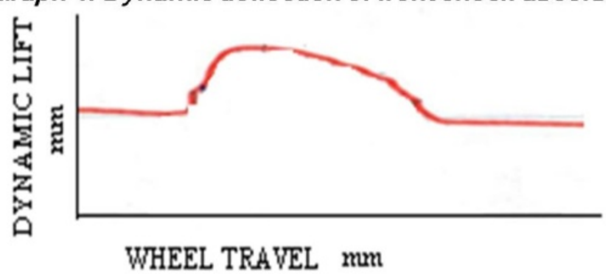
$X_b = X\,[10\,(\theta/\theta_{rise})^3 - 15\,(\theta/\theta_{rise})^4 + 6\,(\theta/\theta_{rise})^5]$ (2)

The eqn. (2) gives the displacement of front axle for any angular position of the wheel affected by the road disturbances and without considering the tyre and suspension deformations. Table 2 based on the eqn. (2) gives the displacement of axle for the maximum amplitude of 25 mm and a semi circular bump giving rise to peak angle of 90^0. Table 2 does not consider the size and contact position of the wheel and compression of the tyre.

Experiment

The assumption is made that the sinusoidal profile of the road is self similar and therefore can be modelled to a smaller size by an eccentric cam (Fig. 10). One fixture of a vertical cam and follower is fabricated in house. The cam diameter is 200 mm with a centre slot of 60 mm for adjusting a desired eccentricity. The eccentricity is adjusted to give the amplitude of sinusoidal road disturbance. Three semicircular bumps of different heights are mounted on cam profile as per the geometric progression of standard ratio of 1.26. The cam is driven by a 3.7 kW geared motor to give a range of revolutions. The follower is a shock absorber in upright position. A platform on the top of shock absorber can take maximum of 100 kg load. The load is 300N and speed is 5.6 m/s. The eccentricity is set at 12.5 mm. and a ball bearing attached to the wheel end of shock absorber for contact with cam.

Graph 4. Dynamic deflection of front shock absorber.



Results and discussions

The graph of the mass end of the shock absorber is recorded on the fixture (Graph 4) and the amplitudes of mass end of the shock absorber are noted as per the parameters set (Table 3). A comparison is made between theoretical and experimental values obtained (Table 4). Based on Table 4, Graph 5 is plotted. The returns are different and jerky. The random profile of return amplitudes does not follow a set pattern for different variation in loads and it appears that the performance of any shock absorber is best for a narrow range of design load and narrow range of design speed. The displacement amplitudes are noted for different loads ranging from 50N to 500N. Different graphs are recorded with similar patterns as in

Graph 5.Dynamic deflection of front shock absorber as per the Table 4

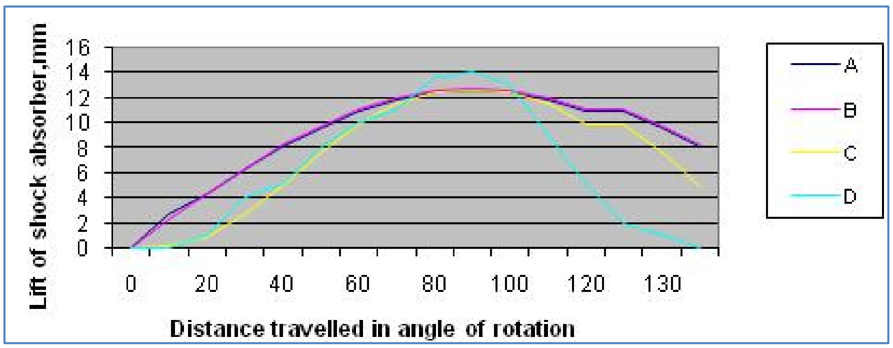


Table 2. Theoretical values of axle lift as per the wheel disturbance due to a road bump.

θ^o	10	20	30	40	50	60	70	80	90
X_b, mm	0.144	0.955	2.62	4.962	7.541	9.875	11.527	12.312	12.5

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Table 3. Measured lift of mass end of shock absorber as per the disturbance due to a bump.

θ^o	10	20	30	40	50	60	70	80	90	100	110	120	121	130	140
X_b, mm	18	19	22	23	26	28	29	31.5	32	31	27	23	20	19	18

Table 4. Comparison of different lifts of shock absorber.

θ^o	10	20	30	40	50	60	70	80	90	100
A	2.170	4.275	6.25	8.034	9.575	10.825	11.746	12.310	12.5	12.310
B	2.212	4.358	6.370	8.189	9.760	11.035	11.973	12.548	12.743	12.548
C	0.144	0.955	2.62	4.962	7.541	9.875	11.527	12.312	12.5	12.312
D	0	1	4	5	8	10	11	13.5	14	13

A= Sinusoidal lift of wheel end of Shock absorber, mm. B = lift of mass end of Shock absorber, as per excitation calculated mm. C= Lift of mass end of Shock absorber. As per polynomial eq. mm. D = Experimental lift of mass end of Shock absorber, mm.

Graph 4 and values of the amplitudes are tabulated in Table 5.

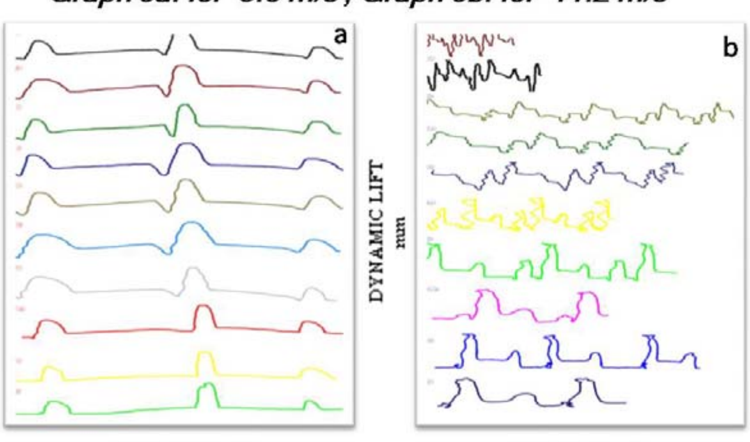
Table 5. Amplitudes taken from displacement graphs.

S. No.	Load, N	Amplitude for bump A, mm	Amplitude For bump B, mm	Amplitude for bump C, mm
1	50	37	23.5	16.7
2	100	39	25	16.5
3	150	49.7	33	29
4	200	26.2	17	43
5	250	27	20.7	40.5
6	300	53	35.5	28.9
7	350	42	27.2	19.7
8	400	56.5	38.5	31.7
9	450	26.7	49	34
10	500	21.5	44.5	27.5

The distress and increased fatigue of the driver caused by the vibrations and dynamic loads result in reduced automobile speed. The average speed reduces by 40-60% and fuel consumption is reduced by 50-70%. Due to the movement of a two wheeler motorcycle over a bump, the driver decelerates to eliminate excessive dynamic loading. The main factor limiting the speed is the shock in the shock absorber. The speed is kept low to assertion the routine behaviour and good road holding. It appears that the low damping passive suspension has a good behaviour in the first part of transient disturbance (rise) but a badly un-damped behaviour in the final part of the transient disturbance (fall). The high damping passive suspension has the complementary behaviour. It is noted that the amplitudes of response is different than the original profile of disturbances.

The maximum delay or the smooth motion is seen to be for 300N load per shock absorber as per the amplitude positions on the graph, while the maximum lift or the disturbance to the load is variable. In all the loading the contact of the absorber and the road i.e. the cam profile is maintained and there is no separation but the maximum lift is more than the height of the bumps mounted on cam. The range of disturbances i.e. difference between maximum and minimum amplitudes is maximum for the highest bump and minimum for the lowest bump. The acceleration for the vertical lift is 0.1 g and is safe for the load of 300N per absorber. The Graph 6a and 6b give the amplitudes of disturbance to different rider loads of 50N, 100N, 150N, 200N, 250N, 300N, 350N, 400N, 450N, and 500N per front shock absorber of a motor cycle. For sufficiently spaced bumps and low speeds the amplitudes of the disturbances are not so violent and vibrational but as the spacing is made closer and as the speed is increased the vibrations increases and causes disturbing vertical amplitudes and hence more vertical acceleration forces.

Graph 6a: for 5.6 m/s ; Graph 6b: for 11.2 m/s



Conclusion

From the results and discussions it is concluded that the amplitude profiles are self similar in nature but the deviation from the regular patterns are due to the transient impulses due to the sudden bumps. For smooth and repetitive curves generated by the wheel on the sinusoidal road profile the polynomial equation is modelled and compared satisfactorily with close approximation with the experimental data.

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