Computer Aided Design for Vehicle Suspension System

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***Abstract*– The comfort level and ride stability of a vehicle are two of the most important factors in a vehicle's subjective evaluation. In the design of a conventional suspension system there is a trade off between the two quantities of ride comfort and vehicle stability. There are two basic types of elements in conventional suspension systems. These elements are springs and dampers. The role of the spring in a vehicle's suspension system is to support the static weight of the vehicle. The role of the damper is to dissipate vibrational energy and control the input from the road that is transmitted to the vehicle.**

This paper introduces the design procedure of coil spring for front independent suspension with double wishbone type suspension and the design procedure of leaf spring for rear dependent suspension. Also, the design calculations of shock absorber for front and rear axle are included. Also, A computer Aided Design (CAD) program for user interface is developed to carry out the design calculations of suspension system.

Keywords—Coil spring design, shock absorber, leaf spring design

I. INTRODUCTION

A considerable research is conducted by many researchers [1-5] to define ride comfort limits. They include shake test, ride simulator experiment. These methods attempt to correlate the response of test subject in terms; discomfort zone and comfort zone with vibrational parameters such as displacement, velocity, and acceleration. For good ride, the natural frequency of the body is around 1 Hz. The evaluation of ride quality of an automobile over a given roadway at constant speed can be investigated. The recommended root mean square, for ride quality is 0 – 0.04 g for smooth rides, 0.04 g – 0.06 g for medium rides and above 0.06 g for rough rides. Often for good ride comfort the suspension system should provide a relatively low vertical stiffness which conflicts with the requirements for a good handling analysis. These conflicting requirements have led to the gradual introduction of independent suspensions, adjustable systems and active elements. Accordingly, analysis of suspension performance in the design stage, using computer simulation is helpful in optimization of the system.

The primary function of the suspension system, therefore, is to isolate the structure, so far as is practicable, from shock loading and vibration due to irregularities of the road surface. Secondly, it must do this without impairing the stability, steering or general handling qualities of the vehicle. The primary requirement is met by the use of flexible elements and dampers, while the second is achieved by controlling, by the use of mechanical linkages, the relative motions between the sprung masses and the unsprung masses. These linkages may be either as simple as a semi elliptic spring and shackle

Modern passenger vehicle usually uses light coil spring. Light commercial vehicles have coil spring at the front and leaf spring at the rear. Heavy commercial vehicle usually uses leaf spring, or air spring to prevent metal to metal contact between the suspension and the frame when the suspension "bottom out" over large bumps or dibs in the road.

The storage energy of suspension spring given in Table 1 depends on:

1) The maximum stress
2) The amount of material at various stress level
3) The modulus of material

TABLE 1: STORAGE OF ENERGY OF SUSPENSION SPRING

Spring type	Energy J/Kg	Maximum stress
Leaf springs	43 - 122	1100 Mpa
Coil springs	510	1100 Mpa

In this paper, the following tasks are investigated:

- The design procedure of suspension system for front coil spring and rear leaf spring.
- The design calculation of the required shock absorber for front and rear axle.
- A computer Aided Design (CAD) program for user interface to carry out the design calculations of vehicle suspension system.

II. DESIGN CALCULATION OF FRONT COIL SPRING FOR INDEPENDENT SUSPENSION SYSTEM

The front suspension is a parallel link independent suspension [2-8] and the geometry in shown in Fig. 1 where the geometry:

- ✓ c =560 mm, a=270 mm

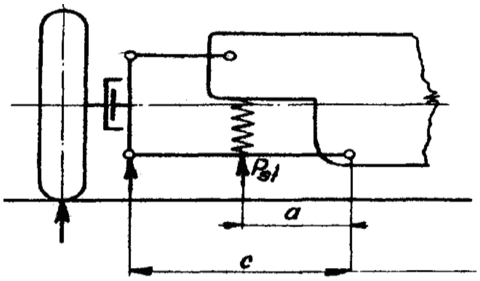

Fig. 1. Parallel link suspension

1, Design calculations of coil spring

The Gross vehicle weight is 37500 N and 50% on front axle

The weight on the one axle

$G_f = \frac{37500}{2} = 18750\ (N)$

Consider the weight transfer is 20% due to declaration
Then the total weight on front axle

G_f =18750 + 0.2x18750 = 22500 (N)

The weight on one wheel

$G_w = \frac{22500}{2}$ =11250 (N)

-

The unsprung mass is about 12%
Then, G_{un}=1350 N

The force acting on each coil spring of independent suspension:

P_{st} =(G_1 - G_{un})$x \frac{c}{a}$

P_{st} =(11250 - 1350)$x \frac{560}{270}$ =20533 (N)

<u>To calculate the required stiffness of coil spring:</u>

Choose:

Torsional stress, [τ] = 450 Mpa

The ratio of mean diameter to wire diameter, $\frac{Dm}{d}$ = 8

The frequency, f = 110 cycle/min for off road vehicle

The angular frequency, ω

$\omega = \frac{2\pi f}{60} = \frac{2\pi * 110}{60} = 11.519$

The static deflection of the spring

$f_{st} = \frac{g}{\omega^2} = \frac{10}{11.915^2} = 0.075\ mm$

The required spring stiffness for coil spring

$k = \frac{W_{wheel}}{f_{st}} = \frac{20533}{0.075} = 273777\ \frac{N}{m} = 273.7\ N/mm$

To calculate the diameter of the wire spring, use this relation:

$\tau * \pi \frac{d^3}{16} = P_{st} * \frac{D_m}{2}$

$d^2 = \frac{64 * P_{st}}{\pi * \tau}$ $\quad d^2 = \frac{64 * 19800}{\pi * 450}$

$d = \sqrt[2]{\frac{64 * 20533}{\pi * 450}}$ $\quad d$ = 30.48 ≃ 31 m

$D_m = 8 * d = 8 * 30.48 = 243.9 \simeq 244\ mm$

-The wire diameter, d=31 mm, and

-The outer diameter, D=244+31=275 mm

- The number of active turns:

$n = \frac{G * d^4}{8 * k * D_m{}^3} = \frac{81.5 * 10^9 * 0.03^4}{8 * 273333 * 0.24^3}$ = 2.2 turn ≃3 turn

The variation of coil spring stiffness with the number of turns is shown in Fig. 2 and is calculated from this relation:

$n = \frac{596923.8}{k}$

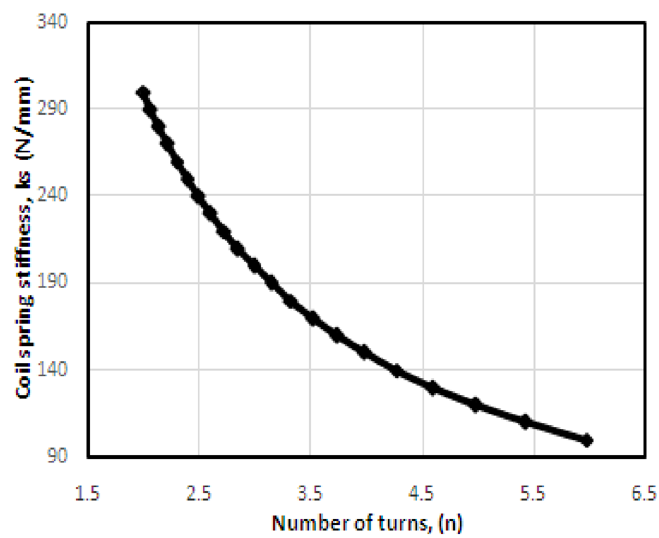

Fig. 2. variations of spring stiffness with the number of turns

<u>Check</u> if τ_{max} doesn't exceed the permissible limiting stress of the material for presupposed δ_{max} using the equation:

$P_{max} \cdot \frac{D}{2} = \frac{\pi * d^3}{16} * \tau_{max}$

$20533 * \frac{0.24}{2} = \frac{\pi * 0.03^3}{16} * \tau_{max}$

$\tau_{max} = 464.77\ Mpa \leq 800\ Mpa$

2. Design calculations of Semi-elliptic Leaf spring

The design calculations of leaf spring is mainly depends on the the load on each spring [2-8]; Fig. 3

1) Design load, P_{st}=10200 N
2) Length between the eye center, L=1320 mm
3) The modulus of elasticity E = 200*10^3 Mpa
4) The stiffening factor SF = 1.4

The required leaf spring stiffness

$$K = \frac{W_{wheel}}{f_{st}} = \frac{10200}{0.075} = 136000\ \frac{N}{m} = 136\ N/mm$$

The spring stiffness, K = 136 N/mm

$$\sum I_1 = \frac{k * L^3}{32 * E * SF} = \frac{136 * 1320^3}{32 * 200 * 10^3 * 1.4}$$

$$\textit{The total moment of inertia}, I_1 = 34910.22\ mm^4$$

Selection of gages: for Width, W = 50 mm

- 3- leaves of 13.2 mm thickness $\sum I = 27006\ mm^4$
- 2- leaves of 9.00 thickness $\sum I = 5798\ mm^4$
- 1- leave of 8.00 thickness $\sum I = 2042\ mm^4$

$$\sum I_2 = 27006 + 5798 + 2042 = 34846\ mm^4$$

$$k_{new} = k_{old} * \frac{\sum I_2}{\sum I_1} = 136 * \frac{34849}{34910}$$

$$k_{new} = 135.775\ mm^4$$

6-leaf spring with width 50 mm is required for each rear suspension.

- The inactive half-length of spring seat 40 mm
- The length at the ends for bedding 25 mm

The length of leaf #6

$$L_6 = 2\left[\frac{I_6}{\sum I}\frac{(L-l)}{2} + l_1 + l_2\right]$$

$$L_6 = 2\left[\frac{2042}{34846} * \frac{(1320-80}{2} + 40 + 25\right]$$

$$L_6 = 202.66\ mm$$

The length of leaf #5

$$L_5 = 2\left[\frac{I_6 + I_5}{\sum I}\frac{(L-l)}{2} + l_1 + l_2\right]$$

$$L_5 = 2\left[\frac{2042 + 2899}{34846} * 620 + 65\right]$$

$$L_5 = 305.8\ mm$$

The length of leaf #4

$$L_4 = 2\left[\frac{I_6 + I_5 + I_4}{\sum I}\frac{(L-l)}{2} + l_1 + l_2\right]$$

$$L_4 = 2\left[\frac{2042 + 2899 * 2}{34846} * 620 + 65\right]$$

$$L_4 = 408.98\ mm$$

The length of leaf #3

$$L_3 = 2\left[\frac{I_6 + I_5 + I_4 + I_3}{\sum I}\frac{(L-l)}{2} + l_1 + l_2\right]$$

$$L_3 = 2\left[\frac{2042 + 2899 * 2 + 9002}{34846} * 620 + 65\right]$$

$$L_3 = 729.32\ mm$$

The length of leaf #2

$$L_2 = 2\left[\frac{I_6 + I_5 + I_4 + I_3 + I_2}{\sum I}\frac{(L-l)}{2} + l_1 + l_2\right]$$

$$L_2 = 2\left[\frac{2042 + 2899 * 3 + 9002 * 2}{34846} * 620 + 65\right]$$

$$L_2 = 1049.6\ mm$$

The length of leaf #1

$$L_1 = 2\left[\frac{I_6 + I_5 + I_4 + I_3 + I_2 + I_1}{\sum I}\frac{(L-l)}{2} + l_1 + l_2\right]$$

$$L_1 = 2\left[\frac{2042 + 2899 * 3 + 9002 * 3}{34846} * 620 + 65\right]$$

$$L_1 = 1370m$$

Length of leave

➢ 1 leaf of 13.2 thickness	I=9002 mm^4
➢ 1 leaf of 13.2 thickness	I=9002 mm^4
➢ 1 leaf of 13.2 thickness	I=9002 mm^4
➢ 1 leaf of 9.00 thickness	I=2899 mm^4
➢ 1 leaf of 9.00 thickness	I=2899 mm^4
➢ 1 leaf of 8.00 thickness	I=2042 mm^4

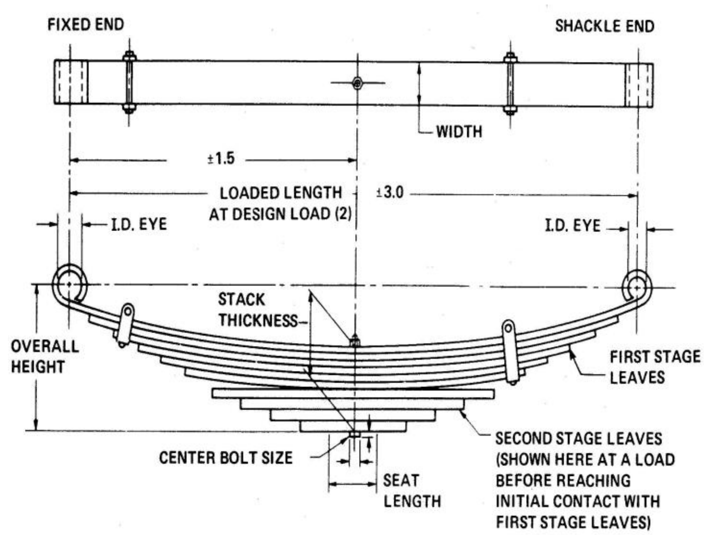

Fig. 3. Rear leaf spring

3. Design calculations of shock absorber

The proper damping ratios are 0.2-0.3. This is appropriate for passenger cars but not enough for race vehicles with higher spring and tire rates, and thus, higher natural frequencies.

Military vehicles require a damping ratio of 0.5-0.7 to control the heave, pitch and roll resonances of the sprung mass [2-6].

To calculate the required damper force, use the damping ratio, ζ is 0.5 for military vehicle, Fig. 4.

<u>Required damping for front axle</u>

P_{st}= 20533 N and k_s=273700 N/m

V_D=0.5 m/sec

$$c = \zeta \times 2\sqrt{m_s k_s}$$

$$c_f = 0.5 \times 2 \times \sqrt{2053 \times 273700}$$

$$c_f = 23704\ N.\sec/m$$

The mean value of front damping force:

$$F_m = V_D \times c_f$$

$$F_m = 0.5 \times 23704\ N$$

$$F_m = 11852\ N$$

let the ratio of rebound force to compression force for front and rear is 4 (q=4)

<u>The compression force:</u>

$$F_c = F_m \frac{2}{1+q} = 11852 \times \frac{2}{1+4} = 4741\ N$$

<u>The rebound force:</u>

$$F_r = F_m \frac{2q}{1+q} = 11852 \times \frac{2 \times 4}{1+4} = 18963\ N$$

<u>Required damping for rear axle</u>

P_{st}=10200 N ks=136000 N/m

V_D=0.5 m/sec

$$c = \zeta \times 2\sqrt{m_s k_s}$$

$$c_r = 0.5 \times 2 \times \sqrt{1020 \times 136000}$$

$$c_r = 11778\ N.\sec/m$$

The mean value of rear damping force:

$$F_m = V_D \times C_r$$

$$F_m = 0.5 \times 11852\ N$$

$$F_m = 5889\ N$$

<u>The compression force:</u>

$$F_c = F_m \frac{2}{1+q} = 5889 \times \frac{2}{1+4} = 2355.6\ N$$

<u>The rebound force:</u>

$$F_r = F_m \frac{2q}{1+q} = 5889 \times \frac{2 \times 4}{1+4} = 9422.4\ N$$

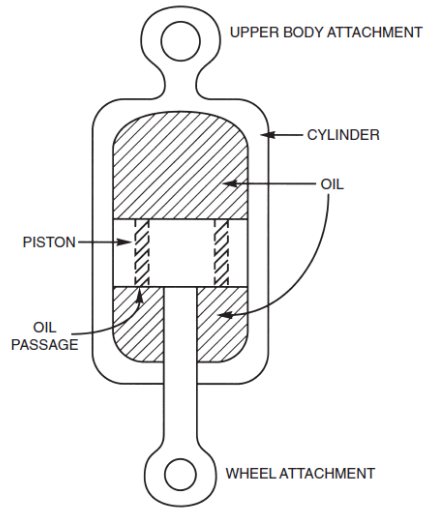

Fig. 4. Shock absorber

III. User Interface

A computer Aided Design (CAD) is developed using python language to carry out the design calculation of independent suspension.. The user must input the Gross vehicle weight. The CAD program will calculate the suspension parameters as shown in Fig. 5

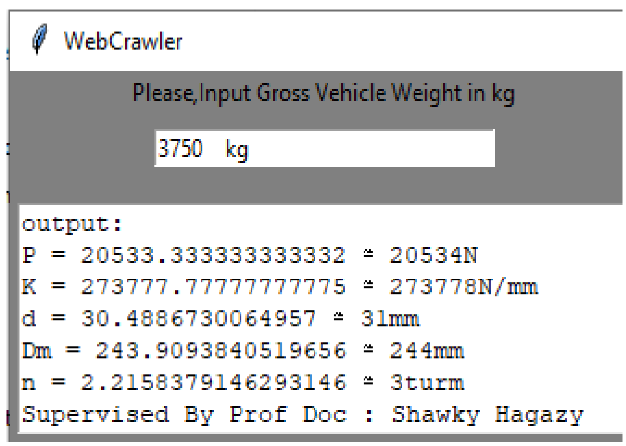

Fig. 5. User interface computer program (CAD)

IV. Conclusions

The paper has been investigated the following:

1. The design procedure of front coil spring for independent suspension.
2. The design procedure of rear leaf spring for dependent suspension.
3. The design procedure of the required shock absorbers for front and rear suspensions.
4. A computer Aided Design (CAD) user interface program is developed using python language to calculate the suspension parameters.

For using the applied load (gross vehicle load), the suspension parameters of system are calculated. The calculated suspensions parameters are compared with the obtained results from the developed program. Both results are identical and same. This gives a trust for the developed interface program for suspension design.

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