Hydraulic Engine Mount Isolation for improved Vehicle Vibration

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The use of hydraulic engine mounts for passenger cars is increasingly becoming common in the industry because they can provide excellent characteristics compared to the conventional elasto-rubber type engine mount. In this paper conception and design strategies for engine mount systems based upon calculation and experimental methods are discussed. The use of the multibody dynamic simulation package ADAMS for Vibration system with rigid or elastic car chassis (Beam system) is discussed.

Introduction

The general vibration isolation system is shown in Fig.1. The "Engine" refers to any structure that creates vibrational disturbance forces. Typical examples are internal combustion engines. The "structure" refers to any structure that is to be isolated from the vibrations of the engine, such as car chassis. Passive mounts are designed to connect two structures and prevent vibrations in one structure from entering into the other . [1]

It is well known that vibration excitation of the engine comes mainly from two sources. One source of the engine vibration is the excitation from road and wheel input acting in vertical direction, and the firing forces at the idling engine speed. The frequency range of this excitation is belower than 30 Hz. At these low frequency vibrations the engine mount should be stiff and highly damped like a shock absorber. The other source is related the unbalanced engine 2nd order forces in the frequency range 25-200 Hz. The engine vibration in this frequency affects strongly the acoustic comfort. To isolate the engine vibrations in this higher frequency range, the engine mount should have a low dynamic stiffness and low damping. [2] Therefore, a major challenge in designing passive mounts, often typified by engine mount, is to make them at the low frequency range as stiff and damply as possible, and dynamically in the high frequency range as soft as possible. Hydraulic engine mounts (HEM) can meet these conflicting performance requirements. Some works have been done on explaining dynamics of hydraulic engine mounts physically and as " Optimum Design Method " mathematically.[3,4]

In this paper the performance design and dynamic response of a hydraulic engine mount are studied in the frequency ranges 0-250 Hz. A new configuration of hydraulic engine mount will be introduced, which may improve vibration isolation properties in the high frequency range. The dynamic ADAMS-Model contains the following characteristics: (1) engine operation forces,(2) the mounted engine block is relatively rigid, (3) the hydraulic engine mounts, (4) the supporting base, a car chassis is rigid or relatively flexible with many degrees of freedom (Beam system). This model is used to conduct a simulation that together with experiments (rigid model) conducted on the mount are used to show the effectiveness of hydraulic engine mounts.

Modeling of Hydraulic Engine Mount

1. Study of the Low frequency range

Fig. 2 illustrates the basic configuration of the HEM studied in the present study. It consists of two chambers with spring, rubber element and a channel for fluid to flow between the chambers. The spring and rubber carry the static and dynamic load on the HEM and act as a piston to pump the fluid through the channel into the lower chamber. The layout of rubber "kidney type " plays an important role in the high frequency range.

Fig.3 presents the mechanical model of the HEM obtained from the actual configuration of the production model of Fig.2. Using the relations of the acting forces under the loads and the continuity condition of fluid, we can derive the following equations from the system. From the continuity equation of fluid can be obtained the fluid volume variation:

$$\dot{V} = A\dot{x} = A_k \dot{x}_k + \kappa \dot{P} \tag{1}$$

Considering the initial conditions $x(0) = x_k(0) = 0$ this equation can be integrated as follows:

$$\begin{aligned} &Ax = A_k x_k + \kappa \cdot (P_t - P_b) \\ &\text{with} \qquad \kappa = \partial V / \partial P \end{aligned} \tag{2}$$

The following equation is formed concerning the fluid in the channel :

$$m_k \{ \ddot{x}_k + \frac{1+\zeta_v}{2l} \, sign(\dot{x}) \dot{x}^2 \} = A_k (P_t - P_b) \tag{3}$$

Under the linearizing conditions the Eq. (3) substitutes with Eq. (3a) :

$$m_k \ddot{x}_k + d_k \dot{x} = A_k (P_t - P_b) \tag{3a}$$

The force F_{iso} transmitted from the HEM to the car chassis is the key parameter of the HEM:

$$F_{iso} = k_s x + d_r \dot{x} + A(P_t - P_b) \tag{4}$$

Consequently, for the engine we have the Eq.(5)

$$M\ddot{x} = F - F_{iso} \tag{5}$$

The transfer function of the force F_{iso} of the displacement x is called the dynamic stiffness. Using the Laplace transformation for Eqs.(1)-(4), the dynamic stiffness is given by Eq. (6)

$$K_{dyn} = \frac{F_{iso}}{x} = k_s + d_r s + K_{hy}\left(1 - \frac{\omega_k^2}{s^2 + 2\xi_k(s)\,\omega_k s + \omega_k^2}\right) \tag{6}$$

where

- A = HEM equivalent fluid piston area
- A_k = HEM mean cross-section area of the channel
- d_r = HEM rubber viscous damping
- F_{iso} = The force transmitted from the HEM to the car chassis - "isolate force"
- F = the excitation force
- k_s = spring + rubber stiffness
- l = the channel length
- M = equivalent engine mass
- m_k = the fluid mass of channel
- P_b = lower chamber fluid pressure of HEM
- P_t = upper chamber fluid pressure of HEM
- x = engine vertical displacement
- x_k = channel fluid displacement
- κ = the compliance of two chambers
- ζ_v = pressure loss coefficient in channel

In Eq. (6) $s = j\omega$ is the Laplace variable, $j = \sqrt{-1}$, $\omega = 2\pi f$ is the vibration frequency. $\xi_k(s)$ is the linearized damping ratio of channel, which is dependent on the frequency. $K_{hy} = A^2/\kappa$ is the hydraulic spring of the HEM .The maximum value for the dynamic stiffness occurs at resonant frequency of the channel ω_k, is equal to

$$\omega_k^2 = \frac{A_k^2}{\kappa \cdot m_k} \tag{7}$$

Under the simplifying conditions $\xi_k = d_r = 0$, the minimum value for the dynamic stiffness occurs at the notch frequency ω_{kn} , is equal to

$$\omega_{kn}^2 = \omega_k^2 \left(\frac{k_s}{k_s + K_{hy}} \right) \tag{8}$$

The dynamic stiffness of no-damping HEM can be defined by following four points

1. static stiffness [$\omega = 0, K_{dyn} = k_s$];

2. minimum stiffness [$\omega = \omega_{kn}, K_{dyn} = 0$];

3. maximum stiffness [$\omega = \omega_k, K_{dyn} = \infty$];

4.stationary stiffness [$\omega = \infty, K_{dyn} = k_s + K_{hy}$].

The phase angle φ is given by Eq. (9).

$$\varphi = \arctan\left(\frac{\mathrm{Im}\,(K_{dyn})}{\mathrm{Re}\,(K_{dyn})} \right) \tag{9}$$

The typical dynamic performance of the HEM can be got by above four points in the Fig.4 as the first evaluated values of designing HEM. Form Fig. 4 ,we see that the dynamic stiffness is more constant at high frequency (f > 20 Hz), which means the fluid of channel is no longer in motion. The channel is "closed". A comparison of the theoretical and experimental dynamic performance is also given by Fig.4. In order to acquire satisfactory dynamic performance of the HEM, the parameters of the HEM such as A, A_k, κ, $\xi_k(s)$ ...... should be studied carefully. For example, the effect of variation of channel area A_k is shown in Fig.5. As A_k is increased, the fluid resonant frequency also increase. The effect of damping $\xi_k(s)$ is considered in the results. The typical changing the damping $\xi_k(s)$ of the HEM is drawn in Fig.6.

2. Optimum Design of the HEM

The model described above appears to be able to predict the dynamic performance of the HEM as well as the effects of changing parameters of the mounts. The criteria for choosing the optimum performance is to minimize the vibration transmitted from the engine through the mount. The ratio of the force F at engine to the force F_{iso} on the chassis of the HEM can be presented in a simplified form as follows:

$$T_R(s)=\frac{F_{iso}(s)}{F(s)}=\frac{F_{iso}(s)}{x(s)}\cdot\frac{x(s)}{F(s)}=K_{dyn}(s)\cdot\frac{x(s)}{F(s)} \qquad (10)$$

This ratio is also called as force transmissibility T_R . A design optimization theory for a shock absorber is applied to the mathematical model of the HEM, which supports the engine mass M Eq.(5).

Eqs (1)-(5) are linearized and non-dimensionalized assuming rubber damping $d_r = 0$ (because little rubber material is necessary for this HEM), channel damping independent on the frequency $\xi_k(s) = \xi_k$ and using the following relationships :

$$G=A/A_k\,,\ \Omega_M=\sqrt{k_s/M}\,,\ \omega_k=\sqrt{K_{hy}/G^2m_k}\,,$$

$$m_\mu=G^2m_k/M,\ \omega_k=\sqrt{K_{hy}/G^2m_k}\,,\ \hat{\omega}=\omega_k/\Omega_M,$$

$$\lambda=\omega/\Omega_M\,,\ K_N=K_{hy}/k_s \qquad (11)$$

The force transmissibility T_R can be expressed as

$$T_R=\frac{-(\hat{\omega}^2m_\mu+1)\cdot\lambda^2+}{\lambda^4-2\hat{\omega}\cdot\xi_k\cdot\lambda^3\cdot j-\{(m_\mu+1)\cdot\hat{\omega}^2+1\}\cdot\lambda^2+}$$

$$\frac{+\,2\hat{\omega}\cdot(1+\hat{\omega}^2m_\mu)\cdot\xi_k\lambda\cdot j+\hat{\omega}^2}{+2\hat{\omega}\cdot(1+\hat{\omega}^2m_\mu)\cdot\xi_k\lambda\cdot j+\hat{\omega}^2} \qquad (12)$$

The optimum parameters for the mass ratio m_μ and for the damping action of the channel ξ_k are determined by Eqs. (13)-(15) . The values used in the calculation are given by the Table 1

$$\hat{\omega}\Big|_{opt}=\frac{\sqrt{2}(K_N+1)}{\sqrt{K_N+2}} \qquad (13)$$

$$\xi_k\Big|_{opt}=\sqrt{\frac{3K_N}{8(K_N+1)}} \qquad (14)$$

$$m_\mu\Big|_{opt}=\frac{G^2m_k}{M}=\frac{K_N}{\left(\hat{\omega}\Big|_{opt}\right)^2} \qquad (15)$$

Symbol	Value	
M	900/g	(kg)
k_s	110	(N/mm)
A	$29^2\,\pi$	(mm^2)
A_K	$4.5^2\,\pi$	(mm^2)
κ	115000	(mm^5/N)
K_{hy}	69.516	(N/mm)
K_N	0.632	(-)
G	36.0	(-)
ϖ	1.423	(-)
	The optimum values	
m_μ	0.312	(-)
ξ_k	0.381	(-)

Table 1 Parameters used in the calculation

Fig.7 shows a comparison of the transmissibility T_R for a conventional rubber mount and above a hydraulic mount on the same engine system.

3. Study of the high frequency range

To meet a low dynamic stiffness and lower damping in the high frequency range, the HEM layout of rubber is configured like a "kidney" so that a second channel of fluid is occurred after the first channel closed. The second channel is called the kidney channel. The fluid mass in the kidney channel is less than the fluid mass in the first channel ($m_n < m_k$). The equivalent area of the kidney channel is greater than the equivalent area of the first channel ($A_n > A_k$). Therefore the resonant frequency of the kidney channel is higher than the resonant frequency of the first channel ($\omega_n > \omega_k$). Hence the dynamic stiffness of the HEM occurs the second notch frequency, which results in a minimum dynamic stiffness in the high frequency range. It is interesting to note that the dynamic stiffness for the HEM operating with kidney channel is still described by equation (3), substituting m_n for m_k , A_n for A_k , x_n for x_k . The overall dynamic stiffness of the HEM is given by Eq.(16), where ω_n and ξ_n are the resonant frequency and damping of the kidney channel respectively. The notch frequency ω_{nn} and the resonant frequency ω_n can be obtained by the same means as studied for the low frequency range .They are expressed by Eqs. (17).

$$K_{dyn} = \frac{F_{iso}}{x} = k_s + d_r\, s + K_{hy}\left(2 - \frac{\omega_k^2}{s^2 + 2\xi_k(s)\,\omega_k\, s + \omega_k^2} - \frac{\omega_n^2}{s^2 + 2\xi_n(s)\,\omega_n\, s + \omega_n^2}\right) \quad (16)$$

$$\omega_n^2 = \frac{A_n^2}{\kappa \cdot m_n}\ ;\ \omega_{nn}^2 = \omega_n^2 \frac{k_s + K_{hy}}{k_s + 2K_{hy}} \quad (17)$$

A_n is a equivalent area of the kidney channel. Thus the dynamic stiffness of no-damping HEM can be defined by two notches and resonant frequencies respectively. In company with above defined four points dynamic stiffness, the other three points of dynamic stiffness can be expressed as follows:

5. second minimum stiffness [$\omega = \omega_{nn}, K_{dyn} = 0$];

6.second maximum stiffness [$\omega = \omega_n, K_{dyn} = \infty$];

7.stationary stiffness [$\omega = \infty, K_{dyn} = k_s + 2K_{hy}$].

It is the something worthy of note that the condition of the fourth point [$\omega = \infty, K_{dyn} = k_s + K_{hy}$] must be modified by [$\omega_k << \omega < \omega_n$, $(k_s + K_{hy}) > K_{dyn} \geq 0$], because the second notch frequency brings the descending changes to the dynamic stiffness until a minimum value. The seven points dynamic performance of no-damping and a comparison of the theoretical and experimental dynamic performance of the "kidney type" HEM are given by Fig.8. The effect of variation in kidney channel high h_n (Fig.3) is to move the second notch (i.g., minimum dynamic stiffness) on the dynamic stiffness curve to the high frequency, see Fig.9. The low dynamic stiffnesses in the high frequency range has been proved to be an utilization for improving interior noise levels throughout the normal driving speed ranges.

ADAMS Model of the HEM System

1. Simulation of Experimental Setup for HEM

An experimental apparatus for the hydraulic engine mount performance has been simulated with the simulation package ADAMS, see Fig.10. This experimental Setup is kind of seismic mechanical vibration generator. The engine mass M is suspended by two linear bearings and moves freely in the vertical direction. This mass includes two flywheels, while in revolution, which rotate in the opposite direction to balance horizontal force. The HEM is mounted between this mass and a basic table. When two flywheels rotate at various rotational speeds, the force dependent on the rotational speed acts at the HEM. Using the data of the Table 1, Fig.11 shows the simulation results for a conventional rubber mount and the HEM in the frequency range 0-30 Hz.

2. Beam Model as car chassis with the HEM

In order to study sound pressure the car chassis is not used as a rigid body, but as a flexible structure, simplified as a Beam System. [5] The data of this model confers with reference [5]. This model consists of a rigid engine, two engine mounts, eleven discrete beam elements, and spring - damping - systems, see Fig. 12. The stiffness of beam is chosen so that the first eigenfrequenz of beam is with in the range 25-30 Hz, the other eigenfrequenz by 70-135 Hz. The simulation is only considered in the vertical direction vibration.

According to great experiences [5], the sound pressure in the car is a function of the element area acceleration. In other words, descending acceleration of the defined element points results in the drop sound pressure. Fig.12 shows the acceleration at the 5th element of the beam for a comparison of the rubber mount and hydraulic mount , when engine excites at 200 Hz. The results indicate that the vibration is improved by the HEM.

Conclusion

In the present study a new configuration of the hydraulic engine mount with "kidney type" has been developed in the low and high frequency range. The design process of the HEM is discussed, which includes (a) Identify the system notch and resonant frequencies, (b) According to optimum conditions determine the parameters necessary to provide dynamic performance. The ADAMS models have been built using the above described formula to achieve a large reduction of vibration transmission at the resonant frequency.

References

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[2] Kyu H.Lee, Young T.Choi, "Performance Design of Hydraulic Mount for Low Frequency Engine Vibration and Noise Control", SAE Paper 941777.

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[6].J. Follmann "Berechnung der dynamischen Kennwerte von Hydrolagern", Diplomarbeit Universität Kaiserslautern, Oktober 1988

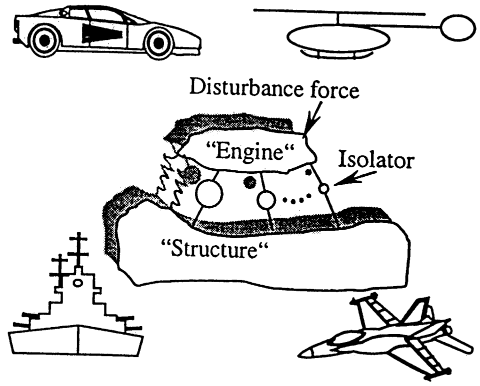

Fig .1 General model of an Isolation System

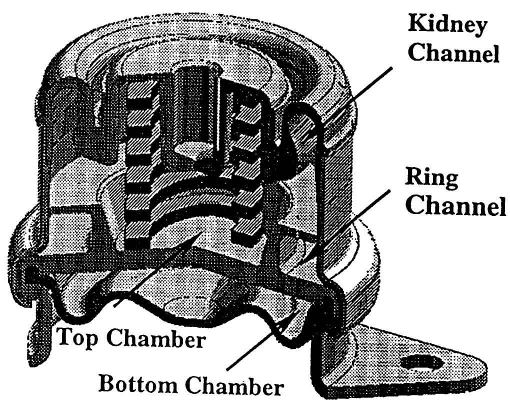

Fig.2 Kidney Type Hydraulic Engine Mount

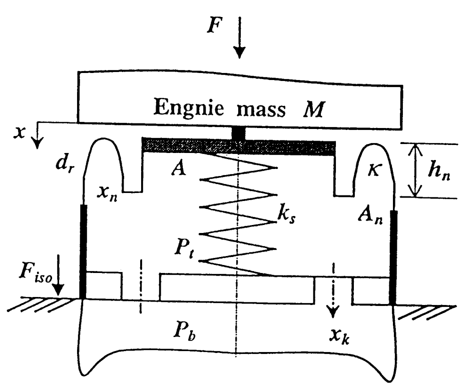

Fig.3 Mechanical Model of the HEM

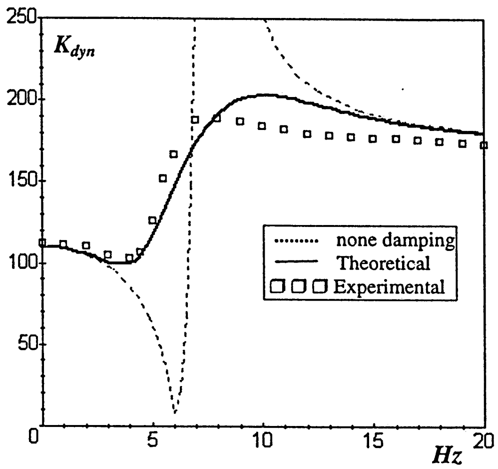

Fig.4 **Dynamic Stiffness of the HEM**

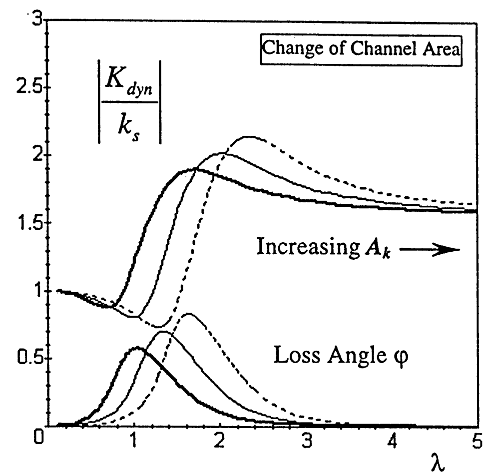

Fig.5 **Dynamic Stiffness of the HEM**

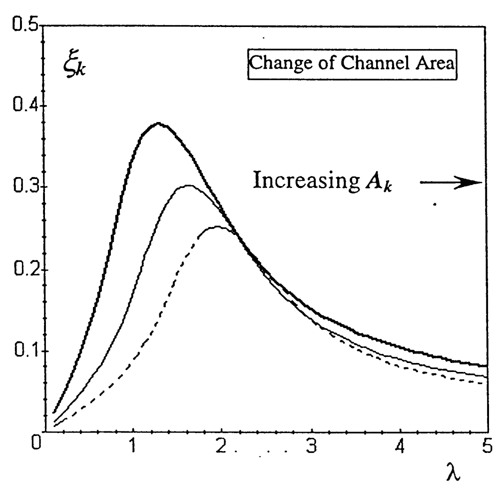

Fig.6 **Typical Damping of the HEM**

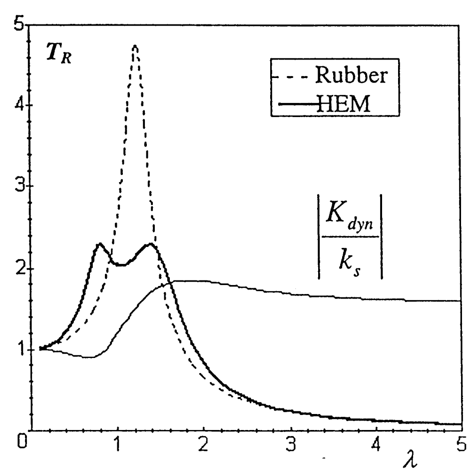

Fig.7 **Transmissbility of HEM and Rubber Mount**

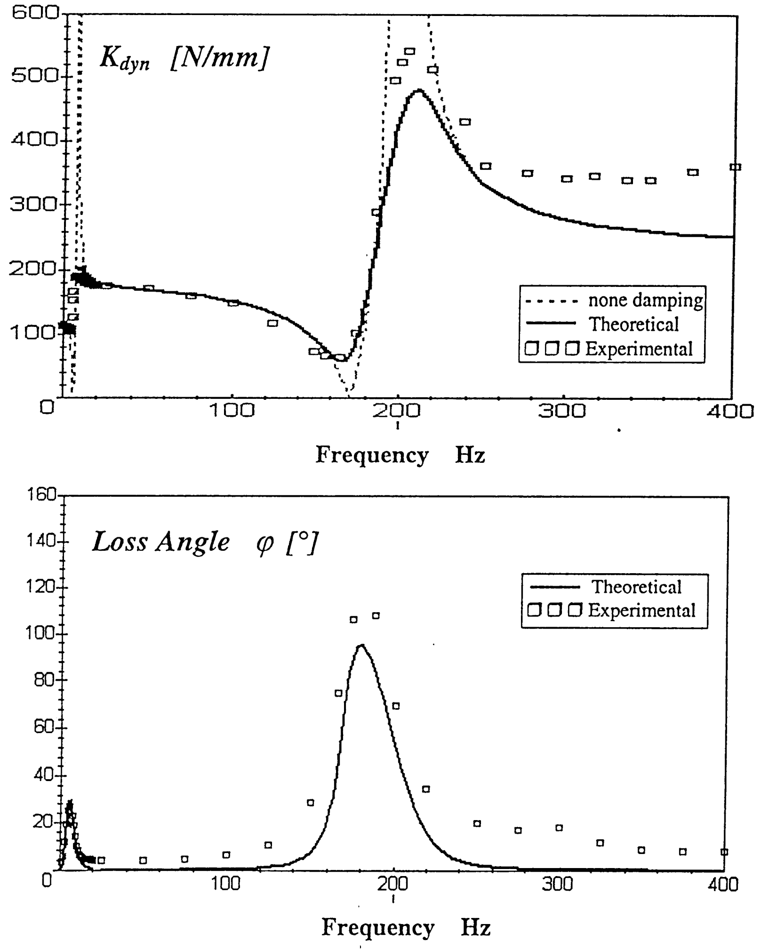

Fig. 8 Comparison of the Experiment and Theoretical Dynamic Stiffness and Loss Angle

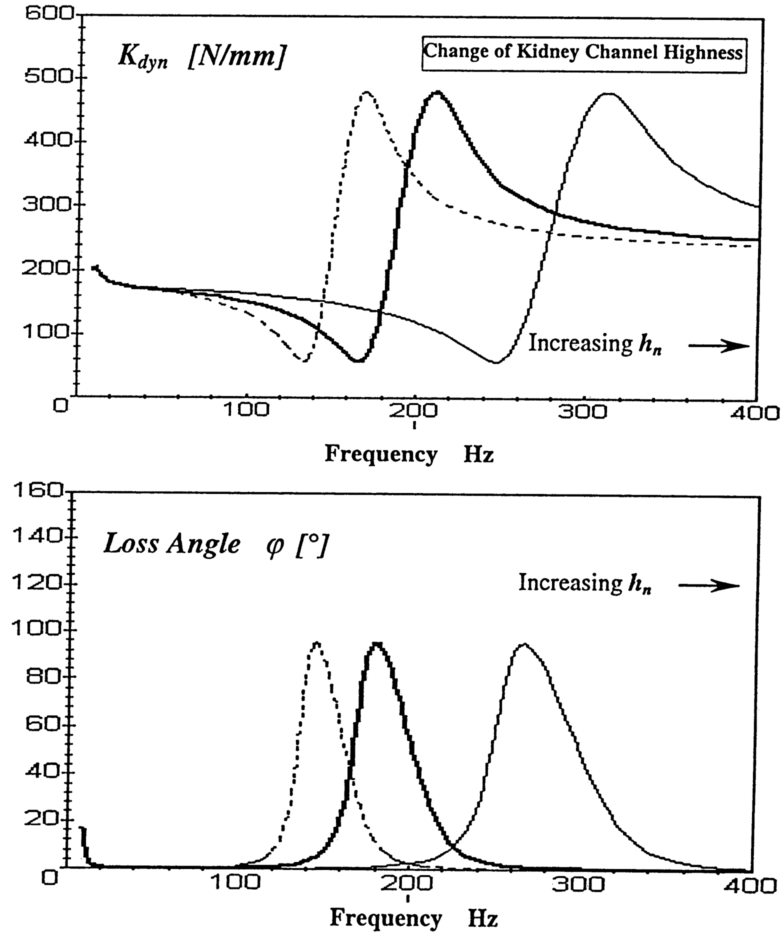

Fig. 9 Effect of the Kidney Channel Highness

<u>Fig. 10</u> ADAMS Simulation of the Experiment Setup

<u>Fig. 11</u> Comparison of the Experiment and Theoretical Dynamic Stiffness & Response

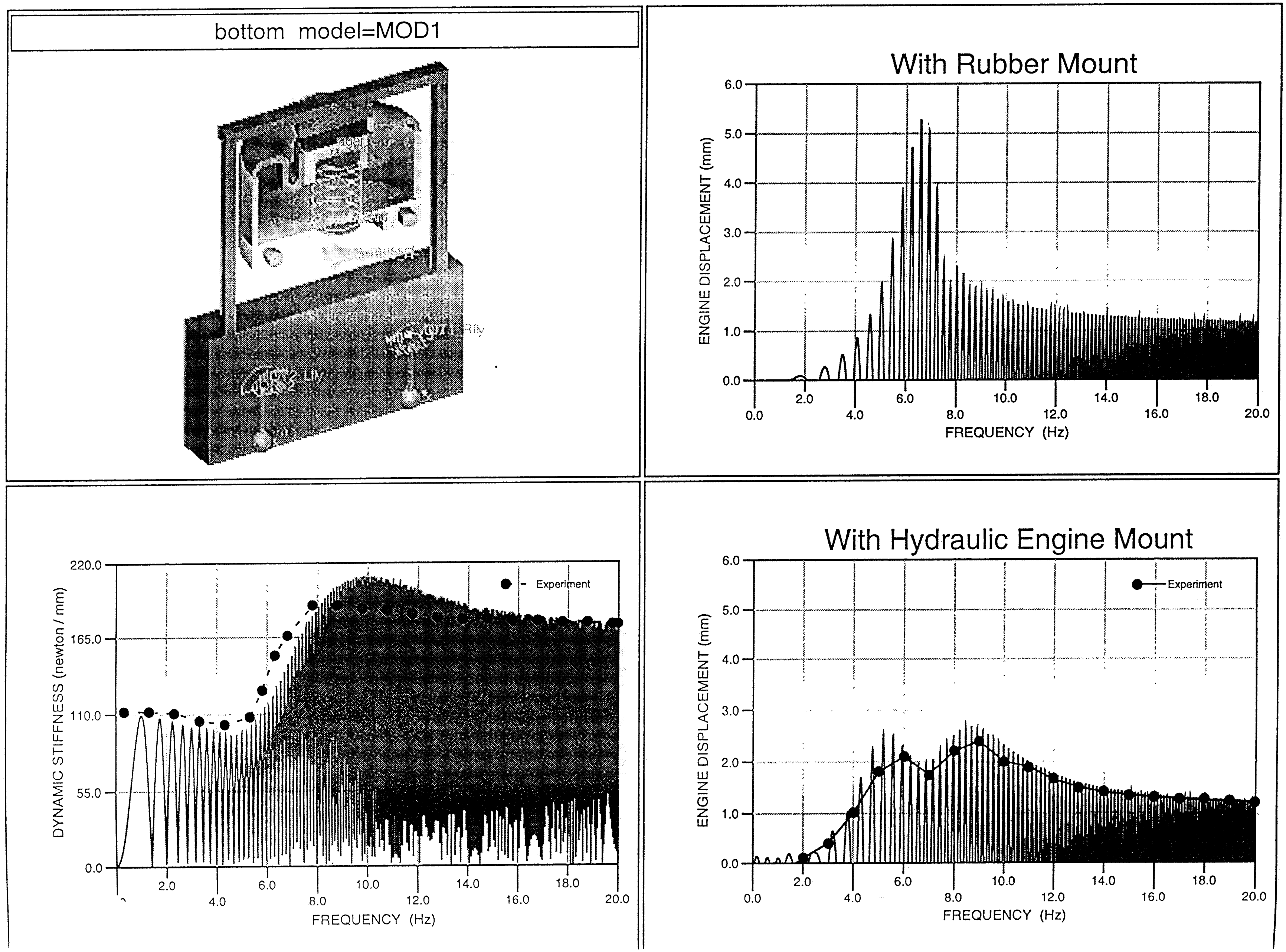

Fig. 12 The acceleration at the 5th element of the beam for a comparison of the rubber mount and hydraulic mount