STUDY OF PASSIVE AND ADAPTIVE
HYDRAULIC ENGINE MOUNTS

DISSERTATION

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By

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To My Parents
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LIST OF SYMBOLS

$A_d$  decoupler area (m$^2$)
$A_i$  cross-sectional area of inertia track (m$^2$)
$A_{i1}, A_{i2}$  cross-sectional areas of inertia track paths 1 and 2 (m$^2$)
$A_p$  equivalent piston area of top element (m$^2$)
$A_t$  cross-sectional area of capillary tube (m$^2$)
a  fluid flow acceleration (m/s$^2$)
b_r  damping coefficient of top element (N·s/m)
b_s  damping coefficient of suspension system (N·s/m)
$C_d$  dynamic discharge coefficient (unitless)
$C_{du}$  ultimate discharge coefficient (unitless)
$C_1, C_{11}, C_{12}$  nonlinear compliances of upper chamber (m$^5$/N)
$C_2, C_{21}, C_{22}$  nonlinear compliances of lower chamber (m$^5$/N)
d_i  inertia track diameter (m)
$F$  excitation engine force (N)
$\bar{F}$  preload (N)
$F_a$  engine force amplitude (N)
$F_T$  transmitted force (N)
$F_{T0}$  mean value of transmitted force (N)
$F_{T1}$  fundamental harmonic of transmitted force (N)
$\hat{F}_{T1}$  amplitude of fundamental harmonic of transmitted force (N)
f  frequency (Hz)
\( f_n \)  natural frequency of inertia track dynamics (Hz)
\( \hat{f}_p \)  peak loss angle frequency (Hz)
\( g \)  gravitational constant (m/s²)
\( h_{is} \)  initial height of liquid column (m)
\( h_o \)  height of liquid column (m)
\( I_d \)  fluidic inertance for decoupler control volume (Kg/m⁴)
\( I_i \)  fluid inertance of inertia track (Kg/m⁴)
\( I_{i1}, I_{i2} \)  fluid inertances of inertia track paths 1 and 2 (Kg/m⁴)
\( I_1 \)  fluid inertance of upper chamber (Kg/m⁴)
\( I_2 \)  fluid inertance of lower chamber (Kg/m⁴)
\( j \)  \( \sqrt{-1} \)
\( K \)  dynamic stiffness modulus (N/m)
\( K' \)  cross-point dynamic stiffness (Ke²j⁰K)
\( K_{e1}, K_{e2} \)  complex spring rate of engine mount (Ke₁+jKe₂)
\( K_{s1}, K_{s2} \)  complex spring rate of suspension system (Ks₁+jKs₂)
\( K_{e1} \)  real part of engine mount complex spring rate (N/m)
\( K_{e2} \)  imaginary part of engine mount complex spring rate (N/m)
\( K_e \)  general nonlinear spring rate of engine mount (N/m)
\( K_{s1} \)  real part of suspension complex spring rate (N/m)
\( K_{s2} \)  imaginary part of suspension complex spring rate (N/m)
\( K_s \)  general nonlinear spring rate of suspension system (N/m)
\( k \)  nominal static stiffness of hydraulic mount (N/m)
\( k_r \)  elastic stiffness of top element (N/m)
\( k_r \)  static stiffness of top element (N/m)
\( k_s \)  elastic stiffness of suspension system (N/m)
\( l_d \) length or height of decoupler control volume (m)
\( l_i \) inertia track length (m)
\( l_{i1}, l_{i2} \) lengths of inertia track paths 1 and 2 (m)
\( m \) index for harmonics
\( m_e \) a portion of engine mass (Kg)
\( m_s \) sprung mass (Kg)
\( n \) number of cycles of excitation stroke
\( \bar{p} \) pressure at static equilibrium condition (N/m²)
\( p_{aim} \) atmospheric pressure (N/m²)
\( p_1, p_{11}, p_{12} \) pressures in upper chamber (N/m²)
\( p_{1\text{max}} \) maximum pressure of upper chamber (N/m²)
\( p_{1\text{min}} \) minimum pressure of upper chamber (N/m²)
\( p_{2}, p_{21}, p_{22} \) pressures in lower chamber (N/m²)
\( Q_{i1}^* \) frequency response function for \( q_{i1} (Q_{i1} e^{j\phi_{q_{i1}}}) \)
\( Q_{i1} \) amplitude of fundamental harmonic of \( q_{i1} \) (m³/s)
\( Q_{i2}^* \) frequency response function for \( q_{i2} (Q_{i2} e^{j\phi_{q_{i2}}}) \)
\( Q_{i2} \) amplitude of fundamental harmonic of \( q_{i2} \) (m³/s)
\( q_d \) volume flow rate through decoupler (m³/s)
\( q_i \) volume flow rate through inertia track (m³/s)
\( q_{i1} \) volume flow rate through inertia track path 1 (m³/s)
\( q_{i2} \) volume flow rate through inertia track path 2 (m³/s)
\( q_1 \) volume flow rate through upper chamber (m³/s)
\( q_2 \) volume flow rate through lower chamber (m³/s)
\( R_d \) nonlinear fluid flow resistance for decoupler (N·s/m⁵)
\( Re \) Reynolds number
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<td>Reynolds numbers for $q_{i1}$ and $q_{i2}$</td>
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<td>$R_i$</td>
<td>nonlinear fluid flow resistance of inertia track (N·s/m$^5$)</td>
</tr>
<tr>
<td>$T_g$</td>
<td>temperature of glycol fluid ($^\circ$C)</td>
</tr>
<tr>
<td>$t$</td>
<td>time (s)</td>
</tr>
<tr>
<td>$V$</td>
<td>fluid volume transfer between upper and lower chambers (m$^3$)</td>
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<tr>
<td>$\bar{V}_{air}$</td>
<td>air volume under static equilibrium (m$^3$)</td>
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<tr>
<td>$V_d$</td>
<td>fluid volume transferred through decoupler (m$^3$)</td>
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<td>$V_{gap}$</td>
<td>decoupler free-volume (m$^3$)</td>
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<td>lower chamber volume increment under static equilibrium from atmospheric condition (m$^3$)</td>
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<td>fluid flow velocity (m/s)</td>
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<td>amplitude of excitation displacement (m)</td>
</tr>
<tr>
<td>$X$</td>
<td>peak value of relative engine displacement (m)</td>
</tr>
<tr>
<td>$X_d$</td>
<td>relative engine displacement or excitation stroke corresponding to $\Delta_d$ (m)</td>
</tr>
<tr>
<td>$\dot{X}_e$</td>
<td>engine acceleration amplitude for fundamental harmonic (m/s$^2$)</td>
</tr>
<tr>
<td>$\ddot{X}_i$</td>
<td>input acceleration amplitude of shaker table (m/s$^2$)</td>
</tr>
</tbody>
</table>
\( \ddot{x}_e \) output acceleration amplitude of engine mass (m/s²)
\( \ddot{x}_s \) sprung mass acceleration amplitude for fundamental harmonic (m/s²)
\( x \) relative displacement or excitation stroke (m)
\( \bar{x} \) displacement under static equilibrium (m)
\( \dot{x} \) excitation velocity (m/s)
\( x_e \) engine displacement (m)
\( x_i \) input displacement of shaker table (m)
\( \ddot{x}_i \) input acceleration of shaker table (m/s²)
\( \ddot{x}_{il} \) fundamental harmonic of \( \ddot{x}_i \) (m/s²)
\( x_o \) output displacement of engine mass (m)
\( \ddot{x}_o \) output acceleration of engine mass (m/s²)
\( \ddot{x}_{o1} \) fundamental harmonic of \( \ddot{x}_o \) (m/s²)
\( x_s \) sprung mass displacement (m)
\( \ddot{x}_e \) engine acceleration (m/s²)
\( \ddot{x}_s \) sprung mass acceleration (m/s²)
\( y \) road profile displacement (m)
\( \alpha_e \) engine acceleration modulus (Kg⁻¹)
\( \alpha_e^* \) engine acceleration (\( \alpha_e e^{i\phi_e} \))
\( \alpha_s \) sprung mass acceleration modulus (Kg⁻¹)
\( \alpha_s^* \) sprung mass acceleration (\( \alpha_s e^{i\phi_s} \))
\( \beta_a, \beta_b \) minor loss coefficients
\( \Delta_d \) decoupler gap (m)
\( \Delta p_f \) pressure drop between lower and upper chambers (N/m²)
\( \Delta p_{fd} \) pressure drop \( \Delta p_f \) related with \( q_d \) (N/m²)
\( \Delta p_{f1} \) pressure drop \( \Delta p_f \) related with \( q_{i1} \) (N/m\(^2\))
\( \Delta p_{f2} \) pressure drop \( \Delta p_f \) related with \( q_{i2} \) (N/m\(^2\))
\( \Delta p_i \) pressure drop through inertia track length (N/m\(^2\))
\( \phi_K \) loss angle or dynamic stiffness phase (deg)
\( \hat{\phi}_x \) peak loss angle (deg)
\( \phi_{qi1} \) phase lead of \( q_{i1} \) with reference to excitation velocity (rad)
\( \phi_{qi2} \) phase lead of \( q_{i2} \) with reference to excitation velocity (rad)
\( \phi_{\alpha_o} \) phase lead of \( \tilde{x}_{\alpha_o} \) with reference to \( \tilde{x}_{\alpha_i} \) (deg)
\( \phi_1 \) phase lead of \( F_{Ti} \) with reference to \( x \) (rad)
\( \nu \) kinematic viscosity of fluid (m\(^2\)/s)
\( \nu_g \) kinematic viscosity of glycol fluid (m\(^2\)/s)
\( \nu_w \) kinematic viscosity of water (m\(^2\)/s)
\( \rho_g \) density of glycol fluid (Kg/m\(^3\))
\( \tau \) time constant (sec)
\( \omega \) circular frequency (rad/s)
\( \omega_n \) natural frequency (rad/s)
\#d \) control volume for decoupler
\#i \) control volume for inertia track
\#1 \) control volume for upper chamber
\#2 \) control volume for lower chamber

**Superscripts**
- \( - \) variables at static equilibrium
- \( * \) variables related to fundamental harmonic
CHAPTER I
INTRODUCTION

1.1 General Background on Mounting Systems

Mounts or suspension systems are indispensable components in a wide variety of dynamic systems such as machinery, equipment, consumer appliances, and vehicles. Their functional characteristics should satisfy various design criteria including vibration isolation, shock absorption, resonance control, noise control, and human comfort. Depending upon the performance characteristics desired, one may employ mechanical (coil springs, elastomers, etc.), fluid (hydraulic or pneumatic), electrical (magnetic or electro-mechanical), or mixed type passive or active mounts. While the linear system theory associated with such mounts is already well established, analysis and design procedures for nonlinear mounts are not well understood. This is because the type of nonlinearities is often specific to the application area and to the given physical device. Therefore, in order to design and control such nonlinear mounts, one must analyze the specific mount from basic engineering principles, and evaluate its dynamic performance in a specific application area. This procedure is followed in this comprehensive study of passive and adaptive hydraulic mounts with application to automotive engines. The example case chosen for this study exhibits nonlinear dynamic characteristics with deflection amplitude, and its mechanical properties (stiffness and damping) are highly frequency-
dependent. Various research issues relevant to the general area of nonlinear dynamics are investigated in detail through analysis, experiment and adaptive system design of an automotive hydraulic engine mount.

1.2 Automotive Engine Mounting Principles

Unbalanced excitation forces inevitably exist in reciprocating automotive engines since they cannot be completely eliminated due to manufacturing tolerances even in multi-cylinder engines. In particular, in automobiles with a large-displacement four cylinder engine, the unbalanced engine forces are primarily due to the engine secondary forces which are produced by crankshaft and connecting rod motion at twice the crankshaft speed (Honda, 1990). For instance, the forcing frequency would be 25-200 Hz for an engine with a speed range from 750 to 6000 rpm. Such unbalanced engine forces mainly act in the vertical direction (Avon, 1987).

The major function of engine mounts, in addition to supporting the engine weight, is to isolate the unbalanced engine forces from the vehicle frame in the frequency range of 25-200 Hz. For this purpose, the engine and gear box unit is usually mounted on three or four engine mounts, and the elastic stiffness of the engine mounts is chosen such that the engine mounting system has a natural frequency of the order of 6-12 Hz in the vertical direction. One cannot use much softer engine mounts to lower the natural frequency further. If the elastic stiffness of engine mounts is too low, the transient response of the engine mounting system can be problematic for the shock excitation. Now, one complicating aspect in the engine mount design is that the engine resonance mode in the frequency
range of 6-12 Hz can be excited by the road profile or wheel excitations and under high-load low-speed conditions: for example, when idling (uneven firing forces at the idling speed) or when the air conditioner is running.

Therefore, two criteria for the specifications of engine mount dynamic properties can be stated as follows. In the frequency range of 6-12 Hz, a higher damping is better for absorbing engine shake or resonances. On the other hand, in the frequency range of 25-200 Hz, a lower dynamic stiffness with a lighter damping is better for low transmission of engine disturbances and especially for reduced noise levels.

Conventional rubber-metal mounts cannot achieve both performance criteria in practice under severe engine compartment conditions where the temperature can rise up to 160°C during vehicle operation. A high-damping rubber mount can yield a low shake level at lower frequencies, but it induces high noise levels at higher frequencies. In addition, the dynamic properties of high-damping elastomers are rather sensitive to temperature changes (Harris and Stevenson, 1987). This aspect may complicate engine mount design since the natural frequency of the engine mounting system can vary with temperature. On the other hand, a low-damping rubber mount yields a low noise level, but it induces high shake levels.

Hydraulic engine mounts have superior dynamic properties to the rubber-metal mounts. The hydraulic mount exhibits high damping coefficient in the idle shake frequency range of 6-12 Hz, whereas it yields low damping coefficient and dynamic stiffness beyond 30 Hz. Therefore, the hydraulic mount functions as a mechanical band pass filter and it can meet both performance criteria, i.e. low peak transmission at resonance and good isolation at higher frequencies. Thus, it provides improved ride
quality and acoustic comfort. Hydraulic mounts are currently being installed in several vehicle models in passive (1992 Toyota Camry) or semi-active (Honda Accord) forms.

1.3 Literature Review on Passive Hydraulic Mounts

Since the early 1980's, various types of passive hydraulic mount have been installed in automobiles to reduce engine vibration and noise (Eberhard and Heitzig, 1984). From the component testing, dynamic properties of the hydraulic mount have been found to be significantly frequency-dependent and highly nonlinear with deflection amplitude (Corcoran and Ticks, 1984). While a few investigators (Shoureshi et al., 1988) have mentioned the nonlinear system parameters of the hydraulic mount, no prior literature exists which rigorously describes the nonlinear dynamics of the hydraulic mount. All of the mathematical models reported in the literature are essentially based on the linear system theory (Flower, 1985; Singh et al., 1992).

This situation is slightly different for another hydraulic damping device that is standard equipment on automobiles, i.e. shock absorber. Lang (1977) and Vliet and Sankar (1983) developed nonlinear mathematical models of automotive and motorcycle shock absorbers respectively. However, the operating frequency range and dynamic characteristics of the hydraulic mount are considerably different from those of the shock absorber. A shock absorber generally works below 10 Hz, whereas the hydraulic engine mount should function effectively up to 100 to 200 Hz. In addition, the shock absorber makes use of sharp-orifice damping, whereas
the damping properties of the hydraulic mount arise from a long orifice or damping channel.

A number of field or laboratory vibration experiments have been carried out to examine the "in-situ" dynamic performance of passive hydraulic mounts. Corcoran and Ticks (1984) conducted field testing for a vehicle installed with two hydraulic mounts and compared their performance with that for rubber-metal mounts. First, the noise level was decreased as much as 5 dBA on a smooth road maneuvering. Second, the acceleration levels at seat tracks were significantly reduced in a simulated bumpy road. Third, the resonance control capability of the hydraulic mounts was shown at the engine-mounting system's natural frequency of 11.3 Hz. Bernuchon (1984) employed hydraulic shakers to excite the vehicle front wheels in the frequency range 2 to 20 Hz with a stroke amplitude of 1.5 mm. Hydraulic mounts decreased the acceleration levels drastically at the seat rail, and also reduced the noise level by 5 to 6 dBB on an open road. Kadomatsu (1989) demonstrated that the bush type hydraulic roll mount could improve the shock isolation characteristics during vehicle acceleration. Seto et al. (1991) excited an engine-mounting system harmonically from 3 to 100 Hz to examine the effect of fluid viscosity on motion transmissibility. Helber and Donker (1990) measured the harmonic, stochastic and transient responses of an engine-mounting system by using a vibration shaker. On the other hand, Seto et al. (1991) and Singh et al. (1992) applied the linear system theory to estimate the motion and force transmissibilities of the engine-hydraulic mounting system respectively.

1.4 Semi-Active and Active Vibration Controls
Passive hydraulic mounts cannot solve all of the problems that arise during vehicle operation. In particular, to resolve some problematic aspects of the passive hydraulic mount and to improve its dynamic performance further, much endeavor has been committed in applying vibration control techniques to the engine mounting area in recent years. Some features of engine-mounting vibration are that it is mainly steady-state behavior and arises under severe environmental conditions.

Engine-mounting vibration control is generally implemented in two forms: semi-active and active vibration controls (Shoureshi et al., 1988). The basic idea of semi-active vibration control is usually to dissipate the vibration energy by changing the engine mount dynamic properties, i.e. damping, with a low-speed and low-power actuator at minimal additional cost. Its scheme is generally implemented in open-loop architecture, and stability problems do not arise since the vibration energy is dissipated. On the other hand, in active vibration control, a counteracting dynamic force is created with dynamic actuator(s) in order to suppress the transmission of the engine disturbance force to the vehicle frame. In other words, an active energy source should be continuously supplied to counteract the continuously-generated target energy source. Its scheme is generally implemented in closed-loop architecture, and, as well-known in the discipline of control engineering, stability problems should always be taken care of in order not to aggravate the vehicle vibration. From the practical viewpoint, semi-active vibration control looks more promising than active vibration control, and may be enough to provide good ride quality for vehicle occupants. Drawbacks of active vibration control are high power consumption and significant hardware cost.
1.5 Literature Survey on Semi-Active and Active Engine Mounts

One example of semi-active vibration control is the adaptive hydraulic mounts. Passive fluid damping is basically utilized to dissipate the engine-mounting vibration energy. The fluid damping is just turned on and off adaptively with a low power actuator, or the fluid damping magnitude is varied in a continuous manner. The on-off adaptive mount employs an electrically operated rotary-solenoid valve (Mizuguchi et al., 1984), a vacuum-operated rotary valve (Honda, 1990), or an electro-dynamic decoupling system (Metzeler and Freudenberg, 1991), all of which are electronically controlled by an on-board microprocessor. Examples of the continuous adaptive mount are to employ a fluid injection system with a hydraulic servovalve (Graf and Shoureshi, 1988), to apply the vacuum in adjusting the fixed-decoupler compliance (Avon, 1987), to use electro-rheological fluid with a high voltage power supply (Duclos, 1987; Ushijima, 1988), and to employ an electrodynamic actuator in the hydraulic working chamber for the vibration compensation system (Metzeler and Freudenberg, 1991).

One example of active vibration control is to apply an electromagnetic actuator as an auxiliary engine mount (Hagino et al., 1986). This active system requires a power amplifier to generate the dynamic force counteracting the engine disturbance force, and a generated force of 30 N was enough to reduce the frame vibration during engine idle. In particular, the electromagnetic actuator should operate normally under hot and harsh engine compartment conditions on a long-term basis. On the other hand, Hodgson (1991) employed the frequency-shaped active control
with a servo-hydraulic system in order to isolate engine disturbance forces from the vehicle frame.

With one exception, all of the adaptive or active engine mounts were basically designed to reduce the lower frequency noise and vibration, especially during engine idle. For instance, Graf and Shoureshi (1988) examined the performance of a prototype semi-active hydraulic mount during engine idle in the speed range from 10 to 20 Hz. However, the adaptive hydraulic mount incorporating an electromagnetic decoupling system in the mount top element (Metzeler and Freudenberg, 1991) was aimed at influencing the high frequency acoustic behavior.

1.6 Scope and Objectives of the Dissertation
The overall objective of this research is to understand the nonlinear dynamic characteristics of the hydraulic mount in depth, and to develop a new adaptive hydraulic mount system that is more effective, efficient, and competitive with broad bandwidth performance features than prior designs.

In-situ vehicle testing is most suitable for comparing the actual performances of any two competing components: for instance, the rubber-metal vs. passive hydraulic mounts. However, for a comprehensive study of the hydraulic mount dynamic characteristics, operator-controllable laboratory testing is more appropriate than vehicle testing. Furthermore, since the hydraulic mount is a highly nonlinear fluid damping device, laboratory vibration testing must be augmented with a nonlinear mathematical analysis. Therefore, one of the main objectives of this study is to develop a simplified analytical model of the passive hydraulic mount
which gives reasonably accurate predictions. By using this mathematical model, we can understand the dynamic behavior of the passive mount both qualitatively and quantitatively, and identify its performance features and limitations in the context of a vehicle model. This should lead to the design of a new adaptive mount. Specific objectives of this study are as follows:

1. To identify, measure and characterize the nonlinear system parameters of the hydraulic mount.
2. To develop and verify a low-frequency (1-50 Hz) mathematical model of the hydraulic mount.
3. To evaluate the performance features both analytically and experimentally, and to identify low-frequency (3-20 Hz) performance limitations of the passive hydraulic mount in a simple vehicle model.
4. To examine the high frequency dynamics (50-250 Hz) of the passive mount by using an experimental methodology.
5. To identify the high-frequency performance problems of the passive mount.
6. To develop a new adaptive hydraulic mount system whose effective bandwidth is much broader than prior designs.

This dissertation is organized into six chapters. The contents of the subsequent chapters are described below.

Chapter 2 introduces the internal configuration of the passive hydraulic mount that is dealt with in this study, along with its pertinent terminology. The excitation frequency-dependent dynamic properties of this hydraulic mount are described on the basis of the component testing results
over 1 to 250 Hz, and the deflection amplitude-dependent nonlinear dynamic properties are identified. The performance limitations at lower frequencies and problems at higher frequencies of the passive mount are identified by performing a quasilinear analysis of the nonlinear vehicle dynamics.

Chapter 3 develops the low-frequency lumped-parameter mathematical model of the hydraulic mount by characterizing, measuring and quantifying its nonlinear parameters such as chamber compliances and flow passage resistances. The mathematical model is verified by comparing predictions with measured responses in both the time and the frequency domains.

Chapter 4 evaluates the dynamic performance of the passive mount rigorously by incorporating its mathematical model formulated in Chapter 3 into a simple vehicle model. Laboratory vibration testing is carried out to augment the vehicle model analysis. Both harmonic (3-20 Hz) and shock responses are examined for the conventional rubber-metal mount and various configurations of the passive hydraulic mount.

Chapter 5 presents a new adaptive hydraulic mount system that employs the engine intake-manifold vacuum for on-off damping control so as to yield broad bandwidth performance features. Since in-situ vehicle setup is not available for this study, only the mechanical actuation scheme and electronic controllers are described in detail without evaluating the new adaptive system further. In particular, microprocessor based on-off valve controller design is illustrated.

Chapter 6 summarizes the results and contributions of this study, and makes recommendation for future work. The appendices contain
manufacturer's specifications for equipment and instrumentation employed in this study, computer codes, and other experimental details.
CHAPTER II

DYNAMIC CHARACTERISTICS OF PASSIVE HYDRAULIC MOUNT

2.1 Introduction

In this chapter, the internal configuration and design features of the passive hydraulic engine mount examined in this study are described. The terminology pertinent to the hydraulic mount and used in describing its dynamic function is defined. The general characteristics of hydraulic mounts are presented on the basis of their dynamic properties measured from component testing. The frequency range of concern is from 1 to 250 Hz. A quasilinear analysis methodology is introduced to evaluate the dynamic performance of the nonlinear hydraulic mount in a vehicle model. In addition, high frequency dynamics of the hydraulic mount are examined by using an experimental methodology.

2.2 Design Features of Hydraulic Mount

From the literature review (Eberhard and Heitzig, 1984; Flower, 1985) and personal discussion with engineers working in the automotive area (Winkler, 1990), it has been found that there exist various types of hydraulic mount and their construction details are not identical. However, in practice, their functional characteristics are not much different from each other. The hydraulic mounts employed in this study are the products of Teledyne Monarch Rubber Co. in Hartville, Ohio. The mount is illustrated in what follows.
2.2.1 Internal Configuration and Operational Features

Figure 2.1 shows the picture of the hydraulic mount cross-section. This mount consists of five parts: the canister, rebound restrictor, top element, lower rubber bellow, and internal subassembly. The same kind of rubber material (duro 51) is used for both the top element and the lower rubber bellow. Basically, the canister is a "can" containing the inside parts such as the top element, lower rubber bellow and internal subassembly, and the rebound restrictor is the "lid of the can". The inherent dynamic characteristics of the hydraulic mount come from the functional interaction of the inside parts which are shown separately in Fig.2.2. Furthermore, the internal subassembly is made up of three components as shown in Fig.2.3: the upper plate or jounce restrictor, the lower plate or decoupler plate, and decoupler. The decoupler is normally produced from the duro 70 rubber.

Now we describe the specific function of each individual part. Figure 2.4(a) depicts the schematic details of the mount. Two, the upper and lower, chambers are filled with the glycol fluid mixture of anti-freeze and distilled water. Note that the mount is equipped with an inertia track and a decoupler. The static stiffness $k_e$ of the top element or upper rubber is selected to support $m_e g$ or $-\bar{F}$, a portion of the engine weight, where $g$ is the gravitational constant and $\bar{F}$ is called the preload. A four-cylinder engine block is generally attached to the vehicle frame through three or four engine mounts. The canister bottom is connected to the vehicle chassis. Since the upper chamber experiences vacuum pressures in conjunction
Figure 2.1  Cross-section of passive hydraulic mount.

Figure 2.2  Inside parts of the hydraulic mount.
Figure 2.3  Disassembled internal subassembly.
with the upward engine motion relative to the chassis, the mount must be assembled in a fluid bath to avoid air entrapment.

A cyclic engine motion causes oscillatory fluid flow between two chambers. A fraction of the displaced fluid is accommodated by the decoupler motion and the remaining portion is forced to flow through the inertia track. The decoupler free travel gap $\Delta_d$ or decoupler gap, in short, is defined in what follows. As illustrated in Fig.2.4(b), $\Delta_d$ is the available space between the jounce restrictor and the decoupler plate within which the decoupler may move freely. Note that the decoupler thickness $t_d$ is typically 2.0 mm for this mount. Provided the inertia track flow is not desired up to a relative engine displacement of $\pm X_d$, $\Delta_d$ is chosen at the design stage as below without considering any dynamic effect.

$$\Delta_d = 2X_d A_p / A_d$$ (2.1)

where $A_p$ is the equivalent piston area of the top element, and $A_d$ is the decoupler area. In the automotive industry, $X_d = 0.2$ mm generally. For this production hydraulic mount with $A_p = 50$ cm$^2$ and $A_d = 23$ cm$^2$, $\Delta_d = 0.7$ mm.

At higher frequencies, the engine is supported only by the low-damping top element along with the very compliant lower rubber bellow since the relative engine displacement generally decreases as the engine speed increases. As a result, the high frequency engine-mounting vibration and the structure-borne noise are isolated from the vehicle chassis. Now, provided that the relative engine displacement amplitude is larger than $X_d$ at lower frequencies, the decoupler sits against the gridwork in the internal subassembly. In consequence, the engine-mounting vibration energy is dissipated by the fluid damping of the inertia track flow. This functional
Figure 2.4 Schematic diagram of the hydraulic mount.
(a) Construction details.
(b) Decoupler and decoupler gap.
characteristic controls the engine resonance amplitude: for instance, during engine idle.

The decoupler enables the hydraulic mount to be amplitude-sensitive, whereas the inertia track renders it to be frequency-dependent. Another favorable aspect of the hydraulic mount is that the effect of temperature on the mechanical properties of the top element can be minimized since it is lightly damped (Harris and Stevenson, 1987).

2.2.2 Design Parameters

Vehicles of different type or model have a different engine mass. As a result, the natural frequencies of their engine-mounting systems, particularly in the vertical direction, may not be identical. This implies that the dynamic properties of a passive hydraulic mount should be tuned to the dynamic characteristics of a specific vehicle in an optimal way before it is installed. As will be explained in Section 3.6.3, the dynamic properties of our hydraulic mount are insensitive to \( \overline{F} \). A most general method of adjusting the mount dynamic properties is to change its inertia track geometry: for instance, the inertia track length \( l_i \) and cross-sectional area \( A_i \).

The flow passage of the inertia track is explained by referring to Fig.2.5(a). When the engine moves downward and the decoupler motion is restricted by the gridwork in the decoupler plate, the displaced fluid in the upper chamber flows into the internal subassembly through hole 1 in the jounce restrictor. It flows counterclockwise along the inertia track and exits the internal subassembly through hole 2 in the decoupler plate. Suppose that there is no fluid flow through the blocking marker.
Conversely, when the engine moves upward and the decoupler motion is restricted by the gridwork in the jounce restrictor, the fluid in the lower chamber flows into hole 2. It flows clockwise along the inertia track and exits the internal subassembly via hole 1. Now, \( l_i \) is defined as the length of the flow passage between hole 1 and hole 2 as denoted in Fig. 2.5(a).

Figure 2.5(b) shows a magnified cross-sectional configuration around the inertia track or section A-A as marked in Fig. 2.5(a). We observe that \( A_i \) is varied by employing different reducer thicknesses. Table 2.1 lists the nominal \( A_i \) and its corresponding inside diameter \( d_i \) in terms of the reducer thickness. Note that "1.0 reducer" represents that the reducer thickness is 1.0 mm, etc. Table 2.2 lists the nominal arc angle for each inertia track code and its nominal \( l_i \) in terms of the inertia track code and the reducer thickness. As an example, \( l_i \) for Code N and 2.0 reducer is 161.3 mm from the table.

Table 2.1 Inertia track cross-sectional area \( A_i \) and its nominal diameter \( d_i \).

<table>
<thead>
<tr>
<th>( A_i ) (mm(^2))</th>
<th>No reducer</th>
<th>1.0 reducer</th>
<th>2.0 reducer</th>
<th>2.5 reducer</th>
<th>3.0 reducer</th>
<th>4.0 reducer</th>
</tr>
</thead>
<tbody>
<tr>
<td>( d_i ) (mm)</td>
<td>7.50</td>
<td>6.75</td>
<td>5.89</td>
<td>5.41</td>
<td>4.91</td>
<td>3.77</td>
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</tbody>
</table>

No reducer: reducer thickness= 0 mm (full cross-sectional area)
1.0 reducer: reducer thickness= 1.0 mm
2.0 reducer: reducer thickness= 2.0 mm
2.5 reducer: reducer thickness= 2.5 mm
3.0 reducer: reducer thickness= 3.0 mm
4.0 reducer: reducer thickness= 4.0 mm
Figure 2.5  Design parameters of hydraulic mount.
(a) Inertia track length.
(b) Inertia track cross-sectional area.
Table 2.2  Inertia track length for each code and reducer.

<table>
<thead>
<tr>
<th>Code</th>
<th>Arc angle (deg)</th>
<th>$l_i$ (mm)</th>
<th>No reducer</th>
<th>1.0 reducer</th>
<th>2.0 reducer</th>
<th>2.5 reducer</th>
<th>3.0 reducer</th>
<th>4.0 reducer</th>
</tr>
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<tr>
<td></td>
<td>angle</td>
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<td></td>
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<tr>
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<td>25.1</td>
<td>24.9</td>
<td>24.8</td>
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<td>35.3</td>
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<td>138.2</td>
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<tr>
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<td>195.6</td>
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<td>191.2</td>
<td>189.8</td>
<td>185.9</td>
<td></td>
</tr>
</tbody>
</table>
2.3 Measurement of Engine Mount Dynamic Properties

In general, the frequency, strain-amplitude, and temperature dependent dynamic properties of engine mounts, or generic mounting components in a broad sense, are represented in terms of the dynamic stiffness spectrum. Figure 2.6 shows how to mathematically determine the transfer dynamic stiffness of mounting components. A sinusoidal displacement excitation \( x \) at frequency \( \omega \) (rad/s) with a given amplitude \( X \) is applied to the mount while \( F_T \), the transmitted dynamic force to the fixed foundation, is obtained. Because engine mounts generally exhibit nonlinear dynamic characteristics, the mean value of \( F_T \) is not necessarily zero and \( F_T \) contains the higher harmonics of \( \omega \) in addition to the fundamental harmonic. Note that the lower harmonics of \( \omega \) are generally negligible.

\[
x(t) = X e^{j\omega t}
\]

\[
F_T(t) = F_{T0} + \sum_{m=1}^{\infty} F_{Tm}(t) = F_{T0} + \sum_{m=1}^{\infty} \tilde{F}_{Tm} e^{j(m\omega t + \phi_m)} = F_{T0} + \sum_{m=1}^{\infty} \tilde{F}_{Tm} e^{j\omega t}
\]

where \( F_{T0} \) is the mean value, \( F_{Tm} (m=1, 2, \ldots, \infty) \) is the harmonics of \( F_T \), \( \tilde{F}_{Tm} \) is the amplitude of \( F_{Tm} \), and \( \phi_m \) is the phase lead of \( F_{Tm} \) with respect to \( x \).

Figure 2.6 Component testing concept.
The complex-valued general cross-point dynamic stiffness $\tilde{K}$ for the given $X$ and at a uniform temperature may be defined as follows:

$$\tilde{K} = \mathcal{F}[F_T(t)]/\mathcal{F}[x(t)]$$

(2.4)

where $\mathcal{F}$ represents the Fourier transformation. Note that $\tilde{K}$ includes the mean component and the higher harmonics of $\omega$ as well as the fundamental harmonic. Now, suppose that the fundamental harmonic $F_{T1}$ of the nonsinusoidal $F_T$ is much greater than its higher harmonics and mean value.

$$F_{T1}(t) = F_{T1}^* e^{j\omega t}$$

(2.5)

where $F_{T1}^*$ is taken to be a complex quantity because in general the phase of $F_{T1}$ will be different from that of $x$. In what follows, the star symbol ($^*$) denotes that the variable is related to the fundamental harmonic. We may define the cross-point dynamic stiffness $K^*(j\omega)$ in terms of the fundamental harmonic as below.

$$K^*(j\omega) = F_{T1}^*/X = K(\omega)e^{j\phi_K(\omega)} = K(\omega)\cos[\phi_K(\omega)] + jK(\omega)\sin[\phi_K(\omega)]$$

(2.6)

where $K$ is the dynamic stiffness modulus and $\phi_K$ is the loss angle representing the phase lead of $F_{T1}$ with reference to $x$. Note that, in industry, the dynamic properties of automotive suspension systems are also measured by applying this component testing concept.

### 2.3.1 Experimental Setup for Component Testing

As illustrated in Fig.2.7, an electrohydraulic material testing system is employed in order to measure $K^*$ of an engine mount. Details of instrumentation, transducers and data acquisition system are given in Section 3.6.1. A compressive preload $\bar{F}$ coinciding with a given $m_e g$ is applied to the mount through the hydraulic actuator. In this study, $\bar{F} = -$
Figure 2.7 Electrohydraulic material testing system for dynamic stiffness measurement.
1200 N unless specified otherwise. The actuator is now excited with a sinusoidal stroke $x$ under closed-loop control.

$$x(t) = X \sin \omega t$$  \hspace{1cm} (2.7)

Usual industrial practice is to apply $X = 1.0 \text{ mm}$ in the frequency range of 1 to 50 Hz and $X = 0.1 \text{ mm}$ over 50 to 250 Hz. The measured $F_T$ is highly nonsinusoidal for the hydraulic mount whereas it is slightly nonsinusoidal for the rubber-metal mount. Now $K$ and $\phi_K$ are estimated on the basis of $F_{T1}$ which is extracted by using the Fourier series analysis or Fourier filter algorithm.

$$F_{T1}(t) = \hat{F}_{T1}(\omega) \sin(\omega t + \phi_1)$$  \hspace{1cm} (2.8)

$$K(\omega) = \frac{\hat{F}_{T1}(\omega)}{X}; \quad \phi_K(\omega) = \phi_1(\omega)$$  \hspace{1cm} (2.9a,b)

where $\hat{F}_{T1}$ is the amplitude of $F_{T1}$ and $\phi_1$ is the phase lead of $F_{T1}$ with reference to $x$.

2.3.2 Typical Dynamic Stiffness Spectra

In what follows, we illustrate the engine mount dynamic stiffness spectra. They were measured in the material testing laboratory at Teledyne Monarch Rubber Co. by employing the high-frequency testing machine, whose experimental set-up is similar to Fig.2.7. A complete list of instrumentation and equipment is given in Appendix A.5. Figure 2.8 shows $K^*(j\omega)$ of the rubber and hydraulic mounts in the frequency range 1 to 250 Hz. The rubber mount coincides with the top element of the hydraulic mount with its fluid drained out.

Figure 2.8(a) shows the typical $K(\omega)$ and $\phi_K(\omega)$ of the low-damping rubber mount. Wave effects are not exhibited at higher frequencies since the length to diameter ratio of the top element is very small. Note that $K$ is
slightly frequency-dependent. Figure 2.8(b) shows the typical $K^*$ of the hydraulic mount with code $N$ at lower frequencies over 1-50 Hz. Its dynamic properties are highly frequency-dependent. The peak loss angle $\hat{\phi}_K$ reaches 50°-60° and decays thereafter. Figure 2.8(c) shows the typical $K^*$ of the hydraulic mount at higher frequencies over 50-250 Hz. Observe the differences from the spectra of the low-damping rubber mount. The hydraulic mount exhibits a reduction in $K$ with a minimum occurring at around 120 Hz, followed by a steep rise thereafter. The level of $\phi_K$ is relatively much higher in the entire frequency range. In addition, $\phi_K > 90°$ over a certain frequency range. It signifies that the high frequency dynamics of the hydraulic mount are different from those of a single degree-of-freedom system; this is explained in more detail in Section 2.6.

2.3.3 Effect of Air Inclusion on Hydraulic Mount Dynamic Properties
A fairly noticeable amount of air was intentionally entrapped into the hydraulic mount during the assembly stage in order to observe its effect on the mount dynamic properties. In fact, we could hear the sound of glycol fluid movement by oscillating the air-entrapped mount by hand; otherwise, we can not hear any sound in general.

Figure 2.9 compares the dynamic stiffness spectra for the hydraulic mounts with and without entrapped air. As shown in Fig.2.9(a), the low frequency dynamic properties are significantly affected by the included air. The levels of both $K$ and $\phi_K$ are reduced particularly in the vicinity of the peak loss angle frequency $\hat{f}_s$, which is around 10 Hz. As a result, the entrapped air may considerably affect the vehicle dynamics at lower frequencies. On the other hand, as shown in Fig.2.9(b), the high-frequency
Figure 2.8  Typical dynamic stiffness spectra.
(a) Low-damping rubber mount.
(b) Hydraulic mount at lower frequencies ($\lambda=1.0$ mm).
(c) Hydraulic mount at higher frequencies ($\lambda=0.1$ mm).
$K'$ are hardly affected by the entrapped air. It is apparent that the included air may almost not affect the vehicle dynamics at higher frequencies.

2.3.4 Effect of Displacement Excitation Amplitude

The nonlinear dynamic characteristics of the hydraulic mount are exemplified by carrying out a series of component testing with various excitation levels. Figures 2.10(a) and (b) show $K'$ of the hydraulic mount excited with $X=1.0$, 2.0 and 3.0 mm. We observe such a deflection amplitude dependent nonlinearity that the levels of $K$ and $\phi_K$ are reduced as $X$ increases. In fact, as illustrated in Fig.2.10(c), $\hat{F}_{T1}$ gets larger with $X$; recall the definition of $K$ given in Eq.(2.9a).

2.4 Quasilinear Analysis of Vehicle Dynamics with Nonlinear Components

In Section 2.3.4, it was shown that the hydraulic mount exhibits nonlinear dynamic properties with deflection amplitude. This implies that the nonlinear mathematical model for the mount should be developed in order to rigorously examine its dynamic performance in a vehicle model. Note that, in the operation of an actual vehicle, the relative engine displacement with respect to the chassis may not be maintained uniform over the frequency range of concern.

In this section, a quasilinear analysis method is presented in order to examine the dynamic performance of the generic engine mount in a simple but reasonable way. The feature of this method is that it incorporates the dynamic stiffness spectra which are obtained from component testing into the vehicle dynamics with nonlinear components such as an engine mount and a suspension system. The basic assumption is that the relative engine
Figure 2.9  Effect of the included air on the dynamic stiffness spectra of hydraulic mount.
(a) Low frequency. (b) High frequency.
Figure 2.10  Variation of dynamic stiffness spectra with displacement excitation amplitude; Code P, No reducer, $\Delta_d=2.24$ mm, duro 90, $t_d=2.0$ mm, $\bar{k}=220$ N/mm.
displacement maintains the excitation level employed in component testing over the frequency range of concern.

In particular, the passive mount has been known to have some problems at higher frequencies, say beyond 150 Hz, probably deteriorating the vibration isolation properties and acoustic behavior of the engine-mounting system. However the high frequency problems have not yet been clarified in a systematic and analytical manner because the fluid dynamics process at high frequencies are not amenable to mathematical modeling. In what follows, the high frequency problems of the on-board passive mount are examined in terms of frequency response functions, i.e. the vehicle chassis accelerance spectra, by employing the quasilinear analysis.

2.4.1 Mathematical Formulation

Figure 2.11 shows the simple vehicle model including an engine mass \( m_e \), engine mount, sprung mass \( m_s \) and suspension system, where \( K_e \) and \( K_s \) represent the frequency, deflection-amplitude and temperature dependent general nonlinear spring rates of the engine mount and suspension system respectively. In addition, \( F(t) \) is the unbalanced or disturbance engine force, and \( x_e(t) \) and \( x_s(t) \) are the dynamic displacements of \( m_e \) and \( m_s \) respectively. The road profile is taken to be uniform; i.e. there is no road profile disturbance input.

The equations of motion for \( m_e \) and \( m_s \) may be written

\[
F(t) - K_e [x_e(t) - x_s(t)] = m_e \ddot{x}_e(t) \tag{2.10}
\]

\[-K_s [x_s(t) - x_e(t)] - K_s x_s(t) = m_s \ddot{x}_s(t) \tag{2.11}\]

In reality, the engine force contains many harmonics and its mathematical formula is quite complicated by itself. However, since the fundamental
harmonic is generally dominant, let us assume that $F$ has only a single harmonic excitation frequency $\omega$. Furthermore, by presuming that the fundamental harmonics of the nonsinusoidal responses $x_e$ and $x_s$ are dominant over the mean values and the higher harmonics, we may write the sinusoidal input and output variables for the vehicle model as follows:

$$F(t) = F_0 e^{j \omega t}$$  \hspace{1cm} (2.12)

$$x_e(t) = X_r^* e^{j \omega t}$$  \hspace{1cm} (2.13)

$$x_s(t) = X_r^* e^{j \omega t}$$  \hspace{1cm} (2.14)

where $F_0$ is the engine force amplitude, and the displacement amplitudes $X_r^*$ and $X_r^*$ are taken to be complex quantities because in general the phases of $x_e$ and $x_s$ will be different from that of $F$. In addition, we consider only the frequency dependent properties for $\kappa_e$ and $\kappa_s$ as below, in terms of fundamental harmonics, by assuming that the deflection-amplitude and temperature take constant values over the frequency range of concern.

$$\kappa_e(j\omega, X, T) = \kappa_e(j\omega) \equiv K_r^*(j\omega)$$  \hspace{1cm} (2.15)

$$\kappa_s(j\omega, X, T) = \kappa_s(j\omega) \equiv K_r^*(j\omega)$$  \hspace{1cm} (2.16)
By substituting Eqs.(2.12-2.16) into Eqs. (2.10) and (2.11), we may obtain the following frequency domain equations:

\[(K_e^* - m_e \omega^2)X_e^* - K_r^* X_r^* = F\]  \hspace{1cm} (2.17)

\[- K_r^* X_r^* + (K_r^* + K_s^* - m_s \omega^2)X_s^* = 0\]  \hspace{1cm} (2.18)

In vehicle dynamics, the occupant ride quality and vehicle maneuverability are correlated with the sprung mass and engine mass acceleration amplitudes $\ddot{x}_r^*$ and $\ddot{x}_s^*$ respectively. They are defined as follows:

\[\ddot{x}_r(t) = -\omega^2 X_r^* e^{j\omega t} = \ddot{x}_r^* e^{j\omega t} \quad ; \quad \ddot{x}_r^* = -\omega^2 X_r^*\]  \hspace{1cm} (2.19a,b)

\[\ddot{x}_s(t) = -\omega^2 X_s^* e^{j\omega t} = \ddot{x}_s^* e^{j\omega t} \quad ; \quad \ddot{x}_s^* = -\omega^2 X_s^*\]  \hspace{1cm} (2.20a,b)

By using the relationships of Eqs.(2.19b) and (2.20b), we may derive the following *accelerance* frequency response functions, from Eqs.(2.17) and (2.18).

\[\frac{\ddot{x}_r^*}{F_s} = \frac{m_e \omega^4 - (K_r^* + K_s^*) \omega^2}{m_e m_s \omega^4 - (m_e K_r^* + m_s K_r^* + m_e K_s^* + m_s K_s^*) \omega^2 + K_r^* K_s^*}\]  \hspace{1cm} (2.21)

\[\frac{\ddot{x}_s^*}{F_s} = \frac{-K_r^* \omega^2}{m_e m_s \omega^4 - (m_e K_r^* + m_s K_r^* + m_e K_s^* + m_s K_s^*) \omega^2 + K_r^* K_s^*}\]  \hspace{1cm} (2.22)

As a result, we are assuming a linear response at each excitation frequency. It follows that the frequency-variant complex spring rates $K_r^*(j\omega)$ and $K_s^*(j\omega)$ may be represented in rectangular form as

\[K_r^*(j\omega) = K_{e1}(\omega) + jK_{e2}(\omega)\]  \hspace{1cm} (2.23)

\[K_s^*(j\omega) = K_{s1}(\omega) + jK_{s2}(\omega)\]  \hspace{1cm} (2.24)

where $K_{e1}$ and $K_{s1}$ are the real parts, and $K_{e2}$ and $K_{s2}$ are the imaginary parts. The real part is associated with the purely elastic properties of the mount or suspension system, whereas the imaginary part is correlated with its damping or energy loss properties. By substituting Eqs.(2.23) and
(2.24) into Eqs.(2.21) and (2.22), we may now define the engine and sprung mass accelerations \( \alpha_e^* \) and \( \alpha_s^* \) as follows:

\[
\alpha_e^*(j\omega) = \alpha_e(\omega)e^{j\phi_e(\omega)} = \frac{\ddot{X}_e^*}{F_a} = \frac{R_{Ne} + jI_{Ne}}{R_D + jI_D}
\]

(2.25)

\[
\alpha_s^*(j\omega) = \alpha_s(\omega)e^{j\phi_s(\omega)} = \frac{\ddot{X}_s^*}{F_a} = \frac{R_{Ns} + jI_{Ns}}{R_D + jI_D}
\]

(2.26)

\[
R_D = m_e m_s \omega^4 - (m_e K_{e1} + m_s K_{s1}) \omega^2 + K_{e1} K_{s1} - K_{e2} K_{s2}
\]

\[
I_D = -(m_e K_{e2} + m_s K_{s2}) \omega^2 + K_{e1} K_{s2} + K_{e2} K_{s1}
\]

\[
R_{Ne} = m_e \omega^4 - (K_{e1} + K_{s1}) \omega^2 ; \quad I_{Ne} = -(K_{e2} + K_{s2}) \omega^2
\]

\[
R_{Ns} = -K_{e1} \omega^2 ; \quad I_{Ns} = -K_{e2} \omega^2
\]

(2.27a-f)

where \( \alpha_e \) and \( \alpha_s \) are the engine and sprung mass acceleration moduli, and \( \phi_e \) and \( \phi_s \) are the phase angles.

Note that \( \alpha_e \) and \( \alpha_s \) are given as follows:

\[
\alpha_e(\omega) = \left| \frac{\ddot{X}_e^*}{F_a} \right| = \sqrt{\left( \frac{R_{Ne}^2 + I_{Ne}^2}{R_D^2 + I_D^2} \right)}
\]

(2.28)

\[
\alpha_s(\omega) = \left| \frac{\ddot{X}_s^*}{F_a} \right| = \sqrt{\left( \frac{R_{Ns}^2 + I_{Ns}^2}{R_D^2 + I_D^2} \right)}
\]

(2.29)

Equations (2.28) and (2.29) state that the acceleration moduli can be determined, provided that the dependence of each complex spring rate on frequency is known.

### 2.4.2 Determination of Complex Spring Rates

Now, the first problem is how to correlate the measured dynamic stiffness spectrum \( K' \) of the engine mount to its complex spring rate \( K'_e \) for examining the vehicle dynamics. Note that the masses of the top element
($m_r=0.3 \text{ Kg}$) and fluid ($m_f=0.1 \text{ Kg}$) of the hydraulic mount are very small compared with $m_e (=122.3 \text{ Kg}$) and $m_s (=270 \text{ Kg}$). Therefore the mechanical impedances of the rubber or hydraulic mounts may be negligible with respect to those of $m_e$ and $m_s$. This is analogous to the concept of the impedance loading effect in electric circuits (Dobelin, 1980). As a result, it is presumed in this quasilinear analysis that $K^*$ determined from the component testing is equivalent to $K_e^*$ in the vehicle model. By comparing Eq.(2.6) with Eq.(2.23), it follows that

$$K_e(\omega)=K(\omega)\cos[\phi_K(\omega)]$$ (2.30)

$$K_{e2}(\omega)=K(\omega)\sin[\phi_K(\omega)]$$ (2.31)

In addition, $x$ coincides with the relative engine displacement with respect to the vehicle chassis or $m_s$.

$$x(t) = x_e(t) - x_s(t)$$ (2.32)

In what follows, $K_e^*$ of the rubber and hydraulic mounts are illustrated. They are calculated on the basis of $K^*$ shown in Fig.2.8 by using Eqs.(2.30) and (2.31). Figure 2.12(a) depicts the slightly frequency-dependent $K_e^*$ of the rubber mount. It is observed that the damping property is relatively insignificant compared to the elastic property. Figure 2.12(b) shows the highly frequency-dependent $K_e^*$ of the hydraulic mount over 1-50 Hz. In particular, the damping property is very significant compared with the rubber mount. Figure 2.12(c) depicts $K_e^*$ of the hydraulic mount over 50-250 Hz. Note that the level of $K_{e2}$ is much higher than that of the rubber mount.

Vehicle suspension systems including shock absorbers are also highly nonlinear; they are frequency-dependent as well as amplitude-sensitive (Lang, 1977). Since the main focus in this study is on the engine
Figure 2.12  Frequency-dependent complex spring rates.
(a) Low-damping rubber mount.
(b) Hydraulic mount at lower frequencies ($X=1.0$ mm).
(c) Hydraulic mount at higher frequencies ($X=0.1$ mm).
mount, the suspension system is represented by a Voigt model with a constant spring stiffness \( k_s \) and a linear viscous damping coefficient \( b_s \). Now we derive the relationships between the complex spring rate and the Voigt model for the suspension system. Applying the component testing concept similar to Fig.2.6, \( F_T \) for this Voigt model may be given as

\[
F_T(t) = F_{T1}(t) = k_s x(t) + b_s \dot{x}(t)
\]  

(2.33)

Substituting Eqs.(2.2) and (2.5) into Eq.(2.33), the cross-point dynamic stiffness is given as

\[
K_s^*(j\omega) = k_s + j b_s \omega
\]

(2.34)

In common with the engine mount, the mechanical impedance of the suspension system is presumed to be negligible in the vehicle model. We obtain \( K_s^* \) from the equivalence of Eq.(2.34) to Eq.(2.24).

\[
K_{s1}(\omega) = k_s \quad ; \quad K_{s2}(\omega) = b_s \omega
\]

(2.35a,b)

### 2.4.3 Result: Accelerance Spectra

Based on \( K_s^*(j\omega) \) as illustrated in Fig.2.12 and \( K_s^*(j\omega) \) as described in the preceding section, we may now calculate \( \alpha_e(\omega) \) and \( \alpha_s(\omega) \) for the vehicle model shown in Fig.2.11 in conjunction with both the rubber and hydraulic mounts. Keep in mind the premise in this quasilinear analysis. The relative engine displacement amplitude \( X \) maintains 1.0 mm in the frequency range 1 to 50 Hz and 0.1 mm over 50 to 250 Hz, no matter what the engine force amplitude is. In other words, \( F_a \) varies at each \( \omega \) to produce the above-mentioned values of \( X \). Other model parameters correspond to a typical medium size passenger car: \( m_e = 122.3 \text{ Kg}, m_s = 270 \text{ Kg}, k_s = 2 \times 10^4 \text{ N/m} \) and \( b_s = 1400 \text{ Ns/m} \).
**Low Frequency**  Figures 2.13(a) and (b) show $\alpha_c(\omega)$ and $\alpha_s(\omega)$ respectively in the frequency range 1 to 50 Hz. The first peak, which occurs around 1 Hz for both the rubber and hydraulic mounts, is due to the resonance mode associated with $k_s$ and $b_s$. In the engine resonance mode which occurs around 9 Hz, we observe excellent resonance control characteristics of the hydraulic mount over the low-damping rubber mount. This results from the higher $K_{e2}$ of the hydraulic mount as mentioned in Fig.2.12(b).

In particular, one performance limitation of the passive hydraulic mount is identified. Once the engine resonance mode is passed, the hydraulic mount yields higher $\alpha_s$ than the rubber mount. This may deteriorate the occupant ride quality, and it results from the upper chamber pressure buildup phenomenon which is explained in detail in Chapter III. Note in Fig.2.12(b) that $K_{e1}$ increases rather abruptly beyond $f_0$ occurring at 10 Hz. It will in turn raise the natural frequency of the vehicle model, and yield higher $\alpha_s$ as a result. An ideally effective antivibration mount should possess a dynamic stiffness that either remains constant or increases slowly with frequency (Snowdon, 1968).

**High Frequency**  Figure 2.13(c) shows $\alpha_s(\omega)$ in the frequency range 50 to 250 Hz. The passive mount yields good, or even better, vibration isolation properties up to 150 Hz like the low-damping rubber mount. However, we observe the problematic aspect of the passive mount beyond 150 Hz; it yields a higher $\alpha_s$ than the low-damping rubber mount. This fact results from the steep increase of $K_{e2}$ with frequency as illustrated in Fig.2.12(c). As a result, the passive mount may deteriorate the isolation properties of the engine mount and in turn cause high noise levels or high "boom". In
Figure 2.13 Accelerance spectra of vehicle model.
actual vehicle testing, the performance of the passive mount has been found to be worse than that of the low-damping rubber-metal mount at higher frequencies (Winkler et al., 1990). In this frequency range, $\alpha_e(\omega)$ exhibits a nearly uniform response coinciding with the asymptotic value as depicted in Fig. 2.13(a). Note from Eq. (2.28) that $\alpha_e(\omega)\approx 1/m_e$ as $\omega \to \infty$.

2.4.4 Effect of Air Inclusion on Accelerance Spectra

We observed in Section 2.3.3 that the entrapped air significantly affects the low frequency dynamic properties of the hydraulic mount as illustrated in Fig.2.9(a). In this section, the corresponding effect on the vehicle dynamics is examined in terms of the vehicle accelerance spectra. Figure 2.14(a) compares the complex spring rates of the hydraulic mounts with and without the entrapped air. In a similar way as for the dynamic stiffness spectra, the contained air makes both $K_{e1}$ and $K_{e2}$ decrease particularly in the vicinity of $f_0$. Now, Fig.2.14(b) compares the vehicle accelerance spectra. The peak value of $\alpha_e$ is -37.4 dB (re 1.0 Kg$^{-1}$) at 19 Hz for the regular mount, whereas it is increased only by 0.4 dB to -37.0 dB (re 1.0 Kg$^{-1}$) at 14 Hz for the air-entrapped mount. On the other hand, the peak value of $\alpha_s$ is -48.0 dB (re 1.0 Kg$^{-1}$) at 16 Hz for the regular mount, whereas it is increased by as much as 2.7 dB to -45.3 dB (re 1.0 Kg$^{-1}$) at 12 Hz for the air-entrapped mount. Note that the tuned frequency is affected by the included air.

In conclusion, the air-entrapped hydraulic mount may deteriorate the occupant ride quality in comparison with the regular mount. Therefore the mount should be assembled in a fluid bath to avoid air inclusion as it is practiced in industry at present.
Figure 2.14  Effect of the included air on the dynamic characteristics of hydraulic mount.
(a) Complex spring rates. (b) Vehicle acceleration spectra.
2.5 Parametric Studies of Hydraulic Mount

In the preceding section, we observed by examining the accelerance spectra of the vehicle model that the passive hydraulic mount is able to suppress the resonant vibration amplitude of the engine-mounting system. In addition, typical design parameters of the hydraulic mount were identified in Section 2.2.2. Now, we examine the effect of various design parameters on the dynamic properties of the passive mount in terms of its dynamic stiffness spectra. The four design parameters accounted for in this section are $A_i$, $l_i$, $\Delta_d$, and the stiffness of the decoupler rubber material which is represented by its durometer value.

A series of component testing on 20 different mounts was carried out for the parametric studies. Table 2.3 lists the mount internal configurations and the test results in terms of $\hat{f}_k$ and $\hat{e}_k$. Figures 2.15, 2.16, 2.17 and 2.18 depict $K^*$ for data sets 1-4, 5-8, 13-16 and 17-20 respectively. The effects of $A_i$, $l_i$, $\Delta_d$ and decoupler stiffness may be identified by comparing the relevant curves in Figs.2.15-2.18. In particular, the effects of $A_i$ and the decoupler stiffness are examined in this section. The effects of $l_i$ and $\Delta_d$ are described in detail in Chapter III, where the measured $K^*$ is compared with the response predicted by the nonlinear mathematical model of the mount. The vehicle accelerance spectra for each data set may be calculated on the basis of its $K^*$ as described in the preceding section.

2.5.1 Effect of Inertia Track Cross-Sectional Area

As an example, let us consider $K^*$ for data sets 1, 2, 13 and 14 illustrated in Figs.2.15 and 2.17. It is apparent that, as $A_i$ decreases from no reducer
Table 2.3 Hydraulic mount configurations for the parametric studies and test results.

<table>
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<th>Channel code</th>
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<th>$\Delta_d$ (mm)</th>
<th>Decoupler duro</th>
<th>$\hat{f}_o$ (Hz)</th>
<th>$\hat{\phi}_X$ (degree)</th>
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<td></td>
<td>4.0</td>
<td>2.25</td>
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Note: all mounts have a nominal static stiffness $k=200$ N/mm.
toward 2.5 reducer and further to 4.0 reducer, \( \hat{\phi}_k \) becomes smaller and the corresponding \( \hat{f}_s \) gets lower.

### 2.5.2 Effect of Decoupler Compliance

Data sets 13-16 and 17-20 are provided to show the decoupler stiffness effect on the mount dynamic properties. Each pair of data sets have the identical configuration except the decoupler stiffness; 90 durometer rubber is very stiff whereas 54 durometer rubber is compliant. Let us compare the corresponding \( K^* \) of each pair from the plots given in Fig.2.17 and 2.18. We observe that the compliant decoupler decreases the levels of both \( K \) and \( \phi_K \).

### 2.5.3 Repeatability of Experiment

Note that the mount configurations for data sets 11 and 12 are identical. Their component testing is performed to examine the repeatability of the measured \( K^* \), as plotted in Fig.2.19. It is obvious that the dynamic properties of the hydraulic mount are considerably repeatable and deterministic.

### 2.6 High Frequency Dynamics of Hydraulic Mount

An experimental methodology is employed in this section to examine and clarify the high frequency dynamics of the hydraulic mount. Recall that, as mentioned in Section 2.4, mathematical modeling of the high frequency dynamics is not attempted in this study. Two sets of component testing are carried out as presented in what follows.

#### 2.6.1 Effect of Excitation Amplitude
Figure 2.15 Dynamic stiffness spectra for data sets 1, 2, 3 and 4.
Figure 2.16  Dynamic stiffness spectra for data sets 5, 6, 7 and 8.
Figure 2.17 Dynamic stiffness spectra for data sets 13, 14, 15 and 16.
Figure 2.18 Dynamic stiffness spectra for data sets 17, 18, 19 and 20.
Figure 2.19  Dynamic stiffness spectra for data sets 11 and 12.
First, Fig. 2.20 shows how $K^*$ varies as $X$ is increased from 0.1 mm to 0.5 mm in a step of 0.1 mm. Since $\Delta_d = 2.24$ mm for the tested mount, $X_d = 0.512$ mm from the relationship given in Eq.(2.1). This implies that the decoupler is maintained to be totally open even for $X = 0.5$ mm. As $X$ varies, both $K$ and $\phi_K$ remain nearly identical up to 50 Hz as in the rubber mount illustrated in Fig.2.8(a). This proves that the decoupler is totally open up to 50 Hz.

We observe one interesting phenomenon beyond 100 Hz. As $X$ is increased, the levels of $K$ and $\phi_K$ are decreased as a whole. In particular, $\phi_K$ is drastically reduced as $X$ changes from 0.3 mm to 0.4 mm. Furthermore, the level of $\phi_K$ is less than 45° for $X = 0.5$ mm. These response characteristics may imply that the oscillating fluid flow inside the upper chamber is more turbulent for the lower amplitude excitation. However, with this set of experiment, we cannot clarify the high frequency fluid dynamics because they are rather obscured by the decoupler dynamics effect. Therefore, we carry out the second set of experiment with a specially prepared hydraulic mount in the following section.

### 2.6.2 Effect of Simple-Orifice Size

Figure 2.21 shows the specially prepared mount. The internal subassembly of the regular mount is replaced with the plain plate having a center hole, which may function as a simple but big orifice. Therefore, let us call this mount the simple-orifice hydraulic mount. In order to clarify the mechanism of the high frequency fluid dynamics, $K^*$ of four different simple-orifice mounts are obtained with each orifice diameter being 20, 30, 40 and 50 mm respectively. Note that the diameter of the decoupler hole for
Figure 2.20 Effect of excitation amplitude on the dynamic stiffness spectra at higher frequencies; Code $P$, No reducer, $\Delta_d=2.24$ mm, duro 90, $\bar{k}=220$ N/mm.
the regular mount is around 54 mm; the gridwork in the decoupler plate and jounce restrictor may reduce the effective area for fluid flow.

Figure 2.22 shows the variations of $K^*(j\omega)$ with the orifice size, where $X=0.1$ mm. In the plots of $K(\omega)$, we observe a consistent pattern. As the orifice gets bigger, the frequency $f_{K_{min}}$ at which the minimum $K$ is exhibited shifts to a higher value and the level of $K(\omega)$ is decreased. It seems like that $f_{K_{min}}$ is a natural frequency of the chamber fluid dynamics. The plots of $\phi_K(\omega)$ introduce two information at $f_{K_{min}}$ for all orifice sizes. First, note that $\phi_K=90^\circ$ at $f_{K_{min}}$; this signifies that something resonates. Second, $\phi_K$ shifts abruptly from about $0^\circ$ toward $180^\circ$ like the phase response of an undamped dynamic system. This indicates that the resonance is not dominated by the fluid flow resistance but rather by the fluid inertial effect.

![Figure 2.21 Hydraulic mount with a simple orifice.](image-url)
Figure 2.22 Effect of orifice size on the high-frequency dynamic stiffness spectra ($\bar{k} = 220\,\text{N/mm}$).
Returning to the plots of $K(\omega)$, we observed that $f_{K_{\text{min}}}$ takes a higher value as the orifice gets bigger. This provides the third but crucial information. As the orifice size increases, the fluid mass $m_2$ over the plate area gets smaller whereas $m_1$ over the orifice becomes bigger; refer to Fig.2.21. Recalling that a natural frequency is inversely proportional to the square root of a certain mass term, it is not the mass related to $m_1$ but to $m_2$ that resonates. The relevant elastic stiffness term may come from fluid compressibility. The upper rubber compliance as measured in Chapter III can not yield such a high resonance frequency.

We now examine the high frequency response in more detail. Figure 2.23 depicts the spectra of $20\log_{10}(K/\bar{k})$ and $\phi_K$ for the 30 mm diameter orifice. Four distinct natural frequencies are observed at about 100, 150, 190 and 250 Hz respectively. Note that $\phi_K=90^\circ$ at each natural frequency. It looks like that the high frequency fluid dynamics are not the kind that can be represented by using a one-dimensional lumped parameter model. Fluid motion is highly turbulent at higher frequencies.

In addition, compare the plots of $K^\prime$ between the 50 mm diameter simple-orifice mount given in Fig.2.22 and the regular mount depicted in Fig.2.8(c). Minor differences may be due to the existence of gridwork as mentioned above and to the decoupler effect of the regular mount. In conclusion, when a passive hydraulic mount is installed in a vehicle, the dynamic stiffness and loss angle rising phenomena at higher frequencies may be inevitable as long as the unbalanced engine force excites the fluid vibration via the top element.

2.7 Discussion
By illustrating the dynamic stiffness spectra measured from component testing, it was shown that the dynamic properties of the hydraulic mount is highly frequency-variant in comparison with the rubber-metal mount. Furthermore, the dynamic stiffness spectrum of the passive hydraulic mount is significantly nonlinear with deflection amplitude. The dynamic stiffness spectrum was employed in the quasilinear analysis of the vehicle dynamics including the nonlinear hydraulic mount. From the quasilinear analysis, we identified both the excellent and problematic aspects of the passive hydraulic mount. As already known, the passive mount provides the engine resonance control as well as vibration isolation characteristics. By varying the mount design parameters such as the inertia track length and cross-sectional area, the mount dynamic properties can be tuned to the engine-mounting resonance frequency of a vehicle. However, two problematic aspects are rather inevitable due to the inherent passive nature of the hydraulic mount. First, at lower frequencies beyond the engine resonance mode, the passive mount yields higher chassis accelerance than the low-damping rubber mount. This results from the stiffening effect of the mount beyond the peak loss angle or damping frequency as obvious from its dynamic stiffness spectrum or complex spring rates. Second, at higher frequencies, the passive mount exhibits worse isolation properties than the rubber mount beyond a certain frequency, *i.e.* 150 Hz. This results from the undesirable side effect of fluid inertia as found from the high-frequency component testing of the simple-orifice hydraulic mounts.

For more accurate analysis of the high-frequency vehicle dynamics, a rigorous mathematical modeling of the high-frequency fluid dynamics in the hydraulic mount is required. However, from a practical viewpoint, the
The ultimate goal of this study is how to avoid the high frequency problems. The adaptive hydraulic mount is presented in Chapter V.

Figure 2.23 Dynamic stiffness spectra for 30 mm diameter orifice.
CHAPTER III
NONLINEAR MATHEMATICAL MODEL OF HYDRAULIC MOUNT

3.1 Introduction

It was identified in the preceding chapter that the hydraulic mount exhibits nonlinear dynamic properties with deflection amplitude. Therefore, the development of the mount nonlinear mathematical model is necessary to rigorously analyze the vehicle dynamics. In this chapter, a lumped-parameter model is formulated for the hydraulic mount dynamics on the basis of fluid system equations. A low frequency nonlinear mathematical model is developed by measuring system parameters and formulating the decoupler switching mechanism. The mathematical model is verified in both the time and frequency domains by comparing the predicted responses with the signals measured in component testing with sinusoidal excitation \( x(t) = X \sin \omega t \). The frequency range of concern is from 1 to 50 Hz. The main objective is to develop a simplified analytical model which gives reasonably accurate predictions.

3.2. Mathematical Formulation

Fluid dynamic characteristics are first modeled with reference to the hydraulic mount having only an inertia track without the decoupler. Its construction details are shown in Fig.3.1(a). The switching dynamics
related to the decoupler function of the regular mount are presented in
Section 3.4.

3.2.1 Rationale for Modeling of Inertia Track Dynamics
A major concern in the analysis of the hydraulic mount dynamics is how to
model the inertia track. The flow velocity in the inertia track is smaller
than the acoustic wave velocity of the glycol fluid (=1500 m/s) by two orders
of magnitude. As a result, fluid density may be taken constant and
thermodynamic effects on the fluid flow may be neglected (Kirshner and
Katz, 1975). At present, the fluidic transmission line theory on the laminar
flow with small-amplitude oscillations is well documented. However the
inherent nature of the inertia track flow is that it is oscillating turbulent
type which is not amenable to accurate theoretical analysis. Therefore, a
lumped-parameter modeling approach looks more appropriate for
representing the inertia track dynamics and clarifying the hydraulic
mount dynamics in a broad sense.

In particular, the fluid circuit for the inertia track has all the fluid
lumped-parameters: resistance, inertance and compliance. The
propagation velocity c through the inertia track is expressed by
\[ c = (1/C_i \rho_g)^{1/2} \]  \hspace{1cm} (3.1)
where \( C_i \) is the fluidic compliance of the inertia track given by the
summation of the fluid and wall compliances. Note that a small clearance
exists in the outer wall of the inertia track between the jounce restrictor and
decoupler plate. This clearance is sealed by the top portion of the lower
rubber bellow. As a result, the wall compliance is dominant over the fluid
compliance.
The acoustic wave length \( \lambda \) for the inertia track at the excitation frequency \( f \) is given by

\[
\lambda = \frac{c}{f}
\]

(3.2)

Suppose that \( c \) is equal to the acoustic wave velocity of the glycol fluid by neglecting the wall compliance. Then, \( \lambda = 30 \text{ m at } f = 50 \text{ Hz} \) and it is larger than the the inertia track diameter by several orders of magnitude. Therefore a wave guide model may not be necessary; one-dimensional flow is assumed. Next, we should determine how many lumps are necessary along \( l_i \) so as to reasonably model its dynamics. As a rule of thumb, 10 lumps per wave length are required for sinusoidal excitation (Doebelin, 1980). Since \( l_i << \lambda / 10 = 3 \text{ m} \), only one lump is employed and the fluidic compliance effect is ignored in this study. However, since \( \lambda \) is reduced when the wall compliance is taken into account, the one-lump model may not be suitable at higher frequencies.

### 3.2.2 Fluid System Equations

Figure 3.1(a) illustrates the lumping scheme to mathematically model the fluid dynamics of the inertia track mount. Note that \#1, \#2 and \#i are the control volumes for the upper chamber, lower chamber and inertia track respectively. Figure 3.1(b) shows the analogous nonlinear lumped-parameter model.

Application of the momentum equation to \#1 and \#2 may yield

\[
p_{12}(t) - p_{11}(t) = I_1(t) \dot{q}_1(t)
\]

(3.3)

\[
p_{22}(t) - p_{21}(t) = I_2(t) \dot{q}_2(t)
\]

(3.4)

where

\( p_{11}, p_{12} = \text{fluid pressures at the ends of } \#1 \ (\text{N/m}^2) \)
Figure 3.1 Hydraulic mount with an inertia track. 
(a) Construction details.
(b) Lumped-parameter model.
\( p_{21}, p_{22} = \) fluid pressures at the ends of \#2 (N/m²)

\( I_1, I_2 = \) time-varying fluid inertances of \#1 and \#2 with the total inertance \( I_1+I_2 \) being constant (Kg/m⁴)

\( q_1, q_2 = \) volume flow rates with regard to \( I_1 \) and \( I_2 \) (m³/s)

In line with the rationale explained in Section 3.2.1, the governing equation for \#i may be derived from the fluid first-order system dynamics as follows:

\[
p_{b}(t) - p_{a}(t) = I_i \dot{q}_i(t) + R_i(\Delta p_i, q_i)q_i(t)
\]  \( (3.5) \)

where

\( p_a, p_b = \) pressures at the ends of \#i (N/m²)

\( I_i = \) time-invariant fluid inertance of \#i (Kg/m⁴)

\( \Delta p_i = p_b - p_a = \) pressure drop (N/m²)

\( q_i = \) volume flow rate through \#i (m³/s)

\( R_i(\Delta p_i, q_i) = \) nonlinear fluid flow resistance inside \#i (N·s/m⁴)

In addition, application of Bernoulli's equation to the sudden contraction and expansion at the inertia track ends yields,

\[
p_{12}(t) + \frac{\rho}{2} \left( \frac{q_i(t)}{A_i} \right)^2 = p_{a}(t) + \frac{\rho}{2} \left( \frac{q_i(t)}{A_i} \right)^2 (1-\beta_a)
\]  \( (3.6) \)

\[
p_{21}(t) + \frac{\rho}{2} \left( \frac{q_i(t)}{A_i} \right)^2 = p_{b}(t) + \frac{\rho}{2} \left( \frac{q_i(t)}{A_i} \right)^2 (1+\beta_b)
\]  \( (3.7) \)

where \( \beta_a \) and \( \beta_b \) are minor loss coefficients associated with abrupt entrance and exit velocities. From Eqs.(3.5-3.7), it follows that

\[
p_{21}(t) - p_{12}(t) = I_i \dot{q}_i(t) + R_i(\Delta p_i, q_i)q_i(t) + \frac{\rho}{2} \left( \frac{q_i(t)}{A_i} \right)^2 (2-\beta_a+\beta_b)
\]  \( (3.8) \)
By representing all of the fluid resistances between $p_{21}$ and $p_{12}$ by $R_I(\Delta \hat{p}_r, q_i)$ with regard to the first-order system dynamics, where the pressure drop $\Delta \hat{p}_r = p_{21} - p_{12}$, Eq.(3.8) may be simplified to

$$p_{21}(t) - p_{12}(t) = I_i \dot{q}_i(t) + R_I(\Delta \hat{p}_r, q_i)q_i(t)$$  \hspace{1cm} (3.9)

Volumetric continuity yields,

$$q_1(t) - A_p \dot{x}(t) = C_{11}(V_{11}, p_{11}) \hat{p}_{11}(t)$$  \hspace{1cm} (3.10)

$$q_i(t) - q_1(t) = C_{12}(V_{12}, p_{12}) \hat{p}_{12}(t)$$  \hspace{1cm} (3.11)

$$q_2(t) - q_i(t) = C_{21}(V_{21}, p_{21}) \hat{p}_{21}(t)$$  \hspace{1cm} (3.12)

$$q_2(t) = C_{22}(V_{22}, p_{22}) \hat{p}_{22}(t)$$  \hspace{1cm} (3.13)

where $C_{11}(V_{11}, p_{11})$ is the nonlinear compliance (m$^3$/N) of the top portion of the upper chamber including the top element, and $C_{22}(V_{22}, p_{22})$ is the nonlinear compliance of the bottom portion of the lower chamber including rubber bellow. Note that the compliance of the metal which the internal subassembly is made up of is negligible in comparison with the fluid compliance. Therefore, $C_{12}(V_{12}, p_{12})$ and $C_{21}(V_{21}, p_{21})$ are nonlinear compliances associated with the fluid volumes just above and below the internal subassembly.

### 3.2.3 Transmitted Force to the Fixed Frame

The dynamic properties of the top element (#r) shown in Fig.3.1 is represented by the Voigt-type model with the frequency-variant elastic stiffness $k_r(\omega)$ and damping coefficient $b_r(\omega)$ in the shear mode. We may derive $k_r(\omega)$ and $b_r(\omega)$ from the measured dynamic stiffness spectra as below.

$$K^*(j \omega) = k_r(\omega) + j b_r(\omega) \cdot \omega = K(\omega) \cos[\phi_K(\omega)] + j K(\omega) \sin[\phi_K(\omega)]$$  \hspace{1cm} (3.14)

$$k_r(\omega) = K(\omega) \cos[\phi_K(\omega)]$$  \hspace{1cm} (3.15a)
\[ b_r(\omega) = K(\omega) \sin[\phi_K(\omega)]/\omega \]  \hspace{1cm} (3.15b)

Figure 3.2 illustrates the typical \( k_r(\omega) \) and \( b_r(\omega) \) in the frequency range 1 to 50 Hz which are calculated from the measured \( K' \) as depicted in Fig.2.8(a).

![Graph showing frequency-dependent mechanical properties of top element.](image)

Figure 3.2 Frequency-dependent mechanical properties of top element.

Figure 3.3 shows the static equilibrium condition (described by an overhead bar) under \( \overline{F} \). Suppose that the cross-sectional areas of the upper and lower chambers are identical to \( A_p \). The transmitted force to the fixed frame is given as

\[ \overline{F}_r = \bar{k}_r \bar{x} + A_p (p_{atm} - \bar{p}) \quad ; \quad \overline{F}_r = \overline{F} \]  \hspace{1cm} (3.16a,b)

where

- \( \bar{k}_r \) = static stiffness of the top element (N/m)
- \( \bar{x} \) = static displacement of the top element (m)
- \( p_{atm} \) = atmospheric pressure (N/m²)
- \( \bar{p} \) = chamber fluid pressure (N/m²)
Equation (3.16b) holds as it should be under static equilibrium. The total transmitted force $F^t_i$ is obtained from Fig.3.4, where the excitation is given as $x^t$.

$$F^t_i(t) = k_r x^t(t) + b_r x^t(t) + A_p (p_{21}(t)-p_{12}(t)-p_{22}(t)+p_{atm})$$  \hspace{1cm} (3.17)

The superscript $t$ denotes the total component including both static and dynamic components.

$$F^t_i(t) = \tilde{F}_i + F_T(t) \quad ; \quad x^t(t) = \tilde{x} + x(t)$$  \hspace{1cm} (3.18a,b)

By using Eq.(3.18) and subtracting Eq.(3.16) from Eq.(3.17), we obtain the (dynamic) transmitted force as follows.

$$F_T(t) = k_r x(t) + b_r \dot{x}(t) + A_p (p_{21}(t)-p_{12}(t)-p_{22}(t) + \tilde{p})$$  \hspace{1cm} (3.19)

### 3.2.4 Low Frequency Model

By making several assumptions on the lumped-parameter governing equations as derived in Sections 3.2.2 and 3.2.3, a low frequency model may be obtained, i.e. in the frequency range 1 to 50 Hz. First, assume that the fluid impedances $j\omega l_1$ and $j\omega l_2$ for the upper and lower chamber inerties are negligible below 50 Hz. As a result, Eqs.(3.3) and (3.4) are simplified to the following: $p_{11}(t) = p_{12}(t) = p_1(t)$ and $p_{21}(t) = p_{22}(t) = p_2(t)$. Second, suppose that $C_{12} = C_{21} = 0$ since $C_{11}$ and $C_{22}$ are more compliant. Consequently, Eqs.(3.11) and (3.12) are approximated to the following: $q_1(t) = q_1(t) = q_2(t)$.

Now denoting $C_1 = C_{11}$ and $C_2 = C_{22}$, we may obtain the low frequency model which is schematically shown in Fig.3.5. Its nonlinear governing equations are derived from Eqs.(3.9),(3.10),(3.13) and (3.19) as follows:

$$p_2(t) - p_1(t) = I_1 \dot{q}_1(t) + R_1 (\Delta p_f q_i) q_i(t)$$  \hspace{1cm} (3.20)

$$q_i(t) - A_p \ddot{x}(t) = C_1(V_1, p_1) \dot{p}_1(t)$$  \hspace{1cm} (3.21)

$$q_i(t) = C_2(V_2, p_2) \dot{p}_2(t)$$  \hspace{1cm} (3.22)
Figure 3.3 Force transmission under static equilibrium.

Figure 3.4 Force transmission under dynamic excitation.
\[ F_T(t) = k_x x(t) + b_x \dot{x}(t) + A_p(\bar{p} \cdot p_1(t)) \]  

(3.23)

where the pressure drop \( \Delta p_f = p_2 - p_1 \).

Figure 3.5 Low frequency mathematical model.

3.3 Measurement of Nonlinear System Parameters

Along with the physical properties of the glycol fluid, \( C_1(V_1, p_1) \), \( C_2(V_2, p_2) \) and \( R_i(\Delta p_f, q_i) \) are measured in this section in order to characterize nonlinear properties of the hydraulic mount. A complete list of instrumentation and equipment is given in Appendices A.1-A.3.

3.3.1 Chamber Compliance

The test setup shown in Fig. 3.6 is employed to measure the pressure vs. volume relationships of upper and lower chambers and to find \( C_1 \) and \( C_2 \). By adjusting the air pressure regulator, chamber pressure is increased by \( \Delta p \). Subsequently, the chamber volume is increased by \( \Delta V \), which is measured by the corresponding decrease in the height of colored-liquid level in the sight glass made up of acrylic rod. Its measuring length is around 9 cm
Figure 3.6  Arrangement for chamber compliance measurement.
(3.54"), and measuring diameters are 1.016 cm (0.4") and 3.048 cm (1.2") for
C₁ and C₂ measurements respectively.

In this experiment, caution must be exercised so as to avoid any
additional compliance effect from being included. Therefore the colored
liquid, acrylic rod and steel tubing are chosen such that their compliances
are negligible in comparison with the rubber chamber compliances C₁ or
C₂. Furthermore, the setup consisting of sight glass, steel tubing and
hydraulic mount block is assembled in a liquid bath to keep the air from
being entrapped.

The measured data for both chambers are shown in Fig.3.7 along
with the compliance measurement concept. Their polynomial relationships
generated by multiple regression routines (Doebelin, 1980) are as follows:

\[ p₁=−6.4V₁+29.2V₁^{7/6}+p_{atm} \]  \hspace{1cm} (3.24)

\[ p₂=5.26×10^{-3}V₂^{2.5}−8.9×10^{-8}V₂^6+1.41×10^{-8}V₂^{6.5}+p_{atm} \]  \hspace{1cm} (3.25)

where \( p₁ \) and \( p₂ \) are absolute pressures (KPa), \( V₁ \) and \( V₂ \) are chamber
volume increments (cc) from the condition that \( p₁=p₂=p_{atm} \). The lower
rubber bellow is more compliant than the upper rubber by an order of
magnitude although both are produced from the same type of rubber
material (duro 51). This is because the lower rubber below is much thinner
than the upper rubber in addition to its geometric effect. Note that \( C₁(V₁,p₁) \)
and \( C₂(V₂,p₂) \) are carefully selected at the manufacturing stage to ensure
that balanced damping forces are produced during both upward and
downward engine motions.

In practice, the experimental setup of Fig.3.6 measures only positive
compliance for \( p₁>p_{atm} \). Now, what will happen when \( p₁<p_{atm} \)? As explained
in Section 3.6.2, \( p₁ \) is always much greater than the vapor pressure of glycol
Figure 3.7 Compliance measurement concepts and measured relationships.
fluid. Therefore it is apparent that some air initially dissolved in the glycol fluid may be released when \( p_1 < p_{\text{atm}} \). In other words, the lower pressure limit does not come from the generation of vapor, but rather from the expansion of gas entrapped in the liquid as \( p_1 \to 0 \). Lang (1977) also observed the release of air initially dissolved in hydraulic oil during the shock absorber experiment.

Consequently, in our mathematical model, \( C_1 \) associated with the released air and the negative compliance of the upper rubber itself is simulated by supposing that a small amount of air volume, say \( V_{\text{air}} = 0.025 \text{ cc} \), is initially present in the top portion of the upper chamber. Note that \( V_1 \) is 40 cc at the static equilibrium under \( F = -1200 \text{ N} \). The negative \( C_1 \) characteristics, calculated on the basis of an ideal gas law, is shown in Fig.3.7(a).

### 3.3.2 Effective Fluid Viscosity

For the hydraulic mount, 50%-50% antifreeze coolant-distilled H\(_2\)O mixture is employed on the basis of the following application criteria: (i) the damping liquid must not freeze during the wintery climate, and (ii) its boiling temperature (typically 160°C) should be higher than that encountered under severe engine compartment conditions. The anti-freeze coolant contains a minimum of 95% glycol mixture generally. Note that the hydraulic oil is not used because it may deform the rubber material and since high temperatures affect its viscosity significantly. In general, viscosity is constant for Newtonian fluids, but it is a function of the shear rate for non-Newtonian fluids (Streeter and Wylie, 1979). Therefore, the
objective of this experiment is to measure the effective viscosity of the glycol fluid in a broad sense.

The effective kinematic viscosity $\nu_g$ of the non-Newtonian glycol fluid may be measured by using a commercially available but expensive liquid viscometer. In our experiment, however, it is measured by employing the step response method as shown in Fig.3.8. This experimental methodology is also employed to examine the effect of the glycol fluid temperature $T_g$ on $\nu_g$. The measurement setup is basically a first-order linear dynamic system. It consists of a column tank containing the liquid of specific weight $\rho_g$ and a linear flow resistance $R_f$ given by three identical laminar-flow capillary tubes; each with length $L_c$ and diameter $D_c$ as given in Fig.3.8 (Doebelin, 1970). The liquid column height $h_o$ during step testing is plotted on an X-Y recorder through a strain-gage type pressure transducer.

Figure 3.9(a) compares the step responses for $H_2O$ (15°C) and glycol fluid (19.4°C and 49°-38°C) in terms of the voltage signal for $h_o$, where the step input or initial column height $h_{is}$ is 50.8 cm (20”).

$$h_o(t)=h_{is}e^{-t/\tau} \quad \tau=A_fR_f/\rho_g$$

(3.26a,b)

where $\tau$ is the time constant and $A_f$ is the cross-sectional area of each capillary tube. Note that the starting points for both fluids do not coincide due to the difference in their densities. The glycol fluid density $\rho_g$ is found to be 1059 Kg/m³.

Now, the semi-logarithmic method is employed to determine the time constant $\tau_g$ of glycol fluid and $\tau_w$ of $H_2O$ as depicted in Fig.3.9(b).

$$Z=ln(h_{is}/h_o) \quad \tau=\Delta t/\Delta Z$$

(3.27a,b)

It is found that $\tau_g=209$ s (19.4°C) and $\tau_w=67.4$ s. Since $R_f$ is proportional to the dynamic viscosity $\mu$, it is apparent that $\tau \propto \nu$. It follows that
Figure 3.8 Arrangement for effective fluid viscosity measurement.
Figure 3.9  Step responses of the fluid first-order system.  
(a) Column height response.  (b) Semi-logarithmic plots.
\begin{equation}
\nu_g = \nu_w \left( \frac{\nu_g}{\nu_w} \right)
\end{equation}

We obtain \(\nu_g = 3.6\) centistokes (3.6 \times 10^{-6} \ m^2/s) at 19.4°C; the kinematic viscosity \(\nu_w\) of H\(_2\)O is 1.16 cSt at 15°C. One feature of this experiment is that the flow rate does not have to be measured since \(\nu_g\) is determined with reference to the well-known \(\nu_w\).

In this step testing, \(h_{is}\) should be as low as possible in order to provide fully developed laminar flow. Note that the transition length \(L_t = 0.0575D_c Re\) for 99% fully developed laminar flow (Merritt, 1967), where \(Re\) is the Reynolds number. It is found that \(L_t = 0.048L_c\) and 0.005\(L_c\) nominally for our H\(_2\)O and glycol fluid systems respectively. As a result, the measured data in this experiment may be reliable.

Now the effect of temperature \(T_g\) on \(\nu_g\) is examined from the step response of the glycol fluid system, where \(T_g\) decreases from 49°C to 38°C during step testing as depicted in Fig.3.9. It is shown that \(\nu_g\) increases as \(T_g\) decreases; for example, \(\nu_g = 2.8\) cSt at 49°C. It follows that the temperature effect is not significant. This may be the most important reason to employ the anti-freeze mixture in hydraulic mounts. The temperature effect on the mount dynamic properties is examined further in the following section.

3.3.3 Fluid Resistance of Inertia Track

Figure 3.10(a) shows the magnified cross-sectional configuration at the blocking marker or section B-B as denoted in Fig.2.5(a). We observe that a "small" clearance exists between the blocking marker on the decoupler plate and the jounce restrictor. Therefore, as illustrated in Fig.3.10(b), the inertia track flow \(q_i\) is actually comprised of \(q_{i1}\) and \(q_{i2}\) through the distinct flow paths \#i\(_1\) and \#i\(_2\) respectively. The length and cross-sectional area of


$i_1$ are denoted by $l_{i1}$ and $A_{i1}$, while $l_{i2}$ and $A_{i2}$ for $i_2$. It is apparent that the pressure drop $\Delta p_{i2}$ related with $q_{i2}$ may be greater than $\Delta p_{i1}$ associated with the major volume flow rate $q_{i1}$, since $i_2$ includes the blocking marker. As a result, $q_{i1}$ and $q_{i2}$ are basically an oscillating turbulent flow through the "long orifice".

In general, the pressure drop $\Delta p_f$ or dynamic discharge coefficient for the oscillating turbulent flow is a function of $A_i/A_p$, $l_i/d_i$, geometrical configuration of the orifice, Reynolds number $Re=\rho g d_i v/l_i$, and acceleration number $c_N=al_i/v^2$, where $v$ and $a$ are flow velocity and acceleration respectively (Lang, 1977). In particular, the dynamic characteristics of both long and short orifices representing the inertia track paths are considerably different from the sharp orifice employed in shock absorbers. Note that, as will be shown in Section 3.6, $q_i$ decreases monotonically beyond a certain resonant frequency ($\sim$10 Hz) of the inertia track fluid inertia. Therefore, the oscillating flow or dynamic effects on $R_i(\Delta p_f,q_i)$ is assumed not to be significant in this study. In practice, considering the present instrumentation technology, the oscillating turbulent flow resistance is rather difficult to measure and incorporate into the lumped-parameter model.

Discharge coefficients for incompressible non-cavitating steady-state flow through long orifices were measured by Lichtarowicz et al. (1965). An empirical formula for the ultimate discharge coefficient $C_{du}$ is given below as a function of the orifice length to diameter ratio $l/d$ in the range from 2 to 10.

$$C_{du}=0.827-0.0085(l/d)$$  \hspace{1cm} (3.29)

where the Reynolds number $Re_h$ based on $\Delta p_f$ was $2 \times 10^4$ in the experiment.
Figure 3.10 Two flow paths of inertia track.
(a) Clearance at the blocking marker.
(b) Two inertia track flows $q_{i_1}$ and $q_{i_2}$. 
\[ Re_h = \frac{d}{\nu} \left( \frac{2\Delta p_{f_1}}{\rho} \right)^{\frac{1}{2}} \]  
(3.30)

Note that \( R_e = C_d Re_h \) when the upstream flow area is much larger than the orifice area. An empirical expression was also developed for the discharge coefficient \( C_d \) in the same range of \( l/d \), but for lower \( Re_h \)'s from 10 to 2\times10^4.

\[ \frac{1}{C_d} = \frac{1}{C_{d_0}} + \frac{20}{Re_h} \left( 1 + 2.25 \frac{l}{d} \right) - \frac{0.005l/d}{1 + 7.5(\log 0.00015 Re_h)^2} \]  
(3.31)

Note that Eqs.(3.29) and (3.31) were derived by employing carefully designed orifices and under nearly ideal flow conditions. However, such ideal conditions may not be met in the automotive hydraulic mount, especially for the path \#i_2 which includes the blocking marker.

Figure 3.11 shows the test setup prepared to measure the discharge coefficients of \#i_1 and \#i_2. The mount block incorporates a take-apart internal subassembly to measure \( \Delta p_{f_1} \) and \( \Delta p_{f_2} \) for various channel codes with a differential pressure transducer. Figure 3.12(a) is a view of the take-apart mount block assembled with a clamping fixture. Figure 3.12(b) shows the pertinent parts of the disassembled mount block: the upper and lower containers, and the internal subassembly. Although it is not clearly shown in Fig.3.12, two rubber rings function as the seal of the mount block along with the clamping fixture. A special adhesive is used to seal the decoupler gap; welding the gridwork in the jounce restrictor or decoupler plate is not recommended. It has been found that the welding process distorts the metal to metal contact between the two plates, and thus inertia track paths are diverted undesirably. Figure 3.13 shows the differential pressure transducer attached to the take-apart mount block through sensing piping.
Figure 3.11 Arrangement for inertia track steady-state fluid resistance measurement.
Figure 3.12  Take-apart mount block used for experimentation.
(a) Overall view assembled with clamping fixture.
(b) Disassembled parts.
Figure 3.13  Assembled differential pressure transducer and take-apart mount block.

Figure 3.14  Hydraulic circuits for turbine meters 1 and 2.
Three different flowmeters are employed to measure $q_i$: turbine flowmeter 1 (operating range=0.19-1.9 lpm), turbine flowmeter 2 (1.9-19 lpm), and rotameter (2.3-23 lpm). Figure 3.14 shows the hydraulic circuits prepared to install turbine flowmeters 1 and 2. A maximum flow rate that the centrifugal water pump can output through turbine meter 2 is around 7.6 lpm, due to a large pressure drop resulting from the small orifice area of this flowmeter. The main reason for employing the centrifugal pump as a flow source is to apply water in evaluating the effect of $T_g$ on $\Delta p_f$, even though this pump is more suitable for low-pressure high-flow rate delivery. Therefore, a variable area flowmeter or rotameter is used for $q_i>7.6$ lpm since its pressure drop is low, and its measurement capability in terms of accuracy (±2.5% full scale) and repeatability (±0.5% full scale) is adequate for this application (Doebelin, 1990). A positive displacement pump such as a gear or vane pump could have been employed for the anti-freeze mixture, even though it is not a petroleum-based liquid. However, long-term usage is not recommended since the anti-freeze mixture contains 50% H$_2$O.

Figure 3.15 shows the measured $\Delta p_f q_i$ relationships of codes A and N with 2.0 reducer at $T_g=40^\circ$C. Here, A1 and N1 correspond to #i$_1$ while A2 and N2 denote #i$_2$. The flow medium is the glycol fluid whose physical properties were described in Section 3.3.2. The nominal total channel length for 2.0 reducer is,

$$l_{i1}+l_{i2}=211.5 \text{ (mm)}$$ \hspace{1cm} (3.32)

Note that Table 2.2 lists the values of $l_{i1}$ in fact, and $l_{i2}$ may be found from Eq.(3.32). In addition, the values of $A_i (=A_{i1}=A_{i2})$ can be obtained from Table 2.1.
As expected, $\Delta p_j$'s for $A2$ and $A1$ comprise the upper and lower bounds. Note that $C_d$ is a constant for potential flow through an ideal orifice, whereas it varies with the Reynolds number or $q_i$ for a long orifice flow as in Eq.(3.31). The meaning of Eq.(3.29) is that $C_d = C_{du}$ for high volume flow rates. We may now employ the multiple regression routines in order to generate the orifice flow relationships from the measured data; the multiple regression and least-squares methods automatically emphasize the region of the high $q_i$.

\[ q_i = C_{du} A_i \left(2\Delta p_f / \rho g \right)^{1/2} \]  

(3.33)

Table 3.1 lists the results along with two values of $C_{du}$ given by Eqs.(3.29) and derived from Eq.(3.33) on the basis of the generated orifice flow relationship, respectively. We observe a close agreement in the values of $C_{du}$ for channels $N1$ and $A1$. The predicted $Re_h$ for $q_i$ are presented in Section 3.6. On the other hand, Eq.(3.33) yields smaller $C_{du}$ for $A2$ and $N2$ than Eq.(3.29) does. This may be related with an additional pressure drop through the blocking marker as mentioned earlier.

<table>
<thead>
<tr>
<th>Code</th>
<th>$l/d_i$</th>
<th>$\Delta p_f$(KPa)-$q_i$(lpm) relationship</th>
<th>$C_{du}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Eq.(3.33)</td>
</tr>
<tr>
<td>$A2$</td>
<td>31.57</td>
<td>$\Delta p_f=2.064q_i^2$</td>
<td>0.310</td>
</tr>
<tr>
<td>$N2$</td>
<td>8.49</td>
<td>$\Delta p_f=0.958q_i^2$</td>
<td>0.455</td>
</tr>
<tr>
<td>$N1$</td>
<td>27.3</td>
<td>$\Delta p_f=0.529q_i^2$</td>
<td>0.612</td>
</tr>
<tr>
<td>$A1$</td>
<td>4.24</td>
<td>$\Delta p_f=0.305q_i^2$</td>
<td>0.806</td>
</tr>
</tbody>
</table>

Table 3.1 Discharge coefficients of various inertia track codes.
Effect of Temperature on Flow Resistance  Now, water is employed as the flow medium in order to examine the effect of $T_g$ on $\Delta p_f$. Note that $v_g$ may ultimately approach unity as $T_g$ is increased. Figure 3.16 compares the pressure drops produced with water and glycol fluid for channel $N1$. Nominal temperature during both experiments is 40°C. We observe that their difference is not significant. For instance, at $q_i=15$ lpm, $\Delta p_f$ for water is only 10% less than that for glycol fluid. Recall from Section 3.3.2 that the effect of $T_g$ on $v_g$ is insignificant. The difference for channel $N2$ may be even smaller since the viscous effect is more noticeable along the channel length; $\beta_a$ and $\beta_b$ are not greatly affected by $v_g$. As a result, it can be concluded that the effect of $T_g$ on the overall dynamic response of the hydraulic mount is minor even under hot engine compartment conditions (say, up to 160°C).

3.4 Mathematical Formulation of Decoupler Dynamics

The mathematical formulation in Section 3.2 and its nonlinear system parameter measurement in Section 3.3 are referred to the inertia track mount. In this section, the switching dynamics of the decoupler is described to complete the mathematical formulation for the regular hydraulic mount.

3.4.1 Lumped-Parameter Model

The control volume $#d$ made to model the decoupler dynamics is shown in Fig.3.17(a). A first-order system may represent the fluid dynamics inside $#d$ as follows:

$$p_2(t)-p_1(t)=I_d q_d(t)+R_d(\Delta p_{fd} q_d(t)q_d(t))$$

(3.34)
Figure 3.15 Measured fluid resistances of various inertia track paths (2.0 reducer).

Figure 3.16 Fluid resistances of channel N1 for glycol fluid and water.
where

\[ I_d = \text{time-invariant fluidic inerstnce for } \#d \text{ (Kg/m}^4) \]

\[ q_d = \text{volume flow rate through } \#d \text{ (m}^3/\text{s}) \]

\[ \Delta p_{fd} = p_2 - p_1 = \text{pressure drop inside } \#d \text{ (N/m}^2) \]

\[ R_d(\Delta p_{fd}, q_d) = \text{nonlinear fluid flow resistance inside } \#d \text{ (N} \cdot \text{s/m}^5) \]

Note that the decoupler compliance is taken zero in this study. As a result, no system parameter measurement is carried out in this formulation.

### 3.4.2 Switching Mechanism for Decoupler Gap

It was described in Section 2.2.1 that the decoupler may move freely between the jounce restrictor and decoupler plate by an amount of \( \Delta_d \). Let us define the decoupler free-volume \( V_{gap} \) as follows:

\[ V_{gap} = A_d \Delta_d \]  \hspace{1cm} (3.35)

In addition, the total fluid volume transferred through the decoupler between the lower and upper chambers is defined as \( V_d \).

\[ V_d(t) = \int_0^t q_d(t) dt \]  \hspace{1cm} (3.36)

Figure 3.17(b) shows the starting transient pressure time histories of the regular mount (\( \Delta_d = 3.0 \text{ mm} \)) at 15 Hz, where \( x = 1.0 \text{ mm} \). Several stages of the switching dynamics associated with the decoupler gap are described with reference to Fig. 3.17(b) in what follows.

**Stage 1** Suppose that the decoupler is centered between the jounce restrictor and decoupler plate at \( t = 0 \). The fluid displaced by the upward \( x \) may begin to flow entirely through the decoupler since \( R_d \) is negligible in comparison with \( R_i \). When \( V_d < 0.5 \ V_{gap} \), the decoupler is open. This is called
Figure 3.17 Decoupler switching dynamics.
(a) Control volume ($#d$) for lumped-parameter model.
(b) Various stages of decoupler dynamics of the regular mount (Code N, 2.0 reducer).
the "decoupled state" since the inertia track is decoupled from the hydraulic mount dynamics. It follows that $p_1=p_2$ and $q_i=0$ in this stage.

**Stage 2** At the moment $V_d = 0.5 V_{gap}$, the decoupler sits under the jounce restrictor. Thus $q_d=0$ thereafter since the decoupler is virtually closed. This is called the "coupled state" since the inertia track is coupled to the mount dynamics. From this moment on till the decoupler is open again, the fluid displaced further by the upward $x$ may flow entirely through the inertia track in the direction from the lower to the upper chambers, *i.e.* $q_i \neq 0$. It follows that $p_1<p_2$ during this stage and $p_1$ is usually lower than $p_{atm}$ (*i.e.* experiencing vacuum) depending on the magnitude of $X$.

**Stage 3** Next, at the moment $p_1 \geq p_2$ in conjunction with the downward $x$, the decoupler is open again, yielding the decoupled state. It follows that $p_1=p_2$ and $q_i=0$ during this stage.

**Stage 4** At the moment $V_d = V_{gap}$, the decoupler sits on the decoupler plate. It is closed again, causing the inertia track flow $q_i$ in the direction from the upper to the lower chambers while $q_d=0$. This is the coupled state and $p_1>p_2$ during this stage.

**Stage 5** Next, at the moment $p_2 \geq p_1$ in conjunction with the upward $x$, the decoupler is open again, yielding the decoupled state. It follows that $p_1=p_2$ and $q_i=0$ during this stage.
**Stage 6**  At the moment $V_d = V_{gap}$, the decoupler sits under the jounce restrictor again. It is closed now, causing the flow $q_i$ in the direction from the lower to the upper chambers while $q_d = 0$. This is the coupled state and $p_2 > p_1$ during this stage.

Thereafter stages 3 to 6 repeat themselves as long as $X$ is large enough to produce the fluid flow through the inertia track as illustrated in Fig. 3.17(b). Note that stages 1 and 2 exist only during the initial transient state. Provided that $x$ is small enough to yield $V_d < V_{gap}$ through the entire period of excitation, the decoupler is never closed at any moment and $q_i = 0$.

### 3.5 Development of Nonlinear Mathematical Model

A low frequency mathematical model is now developed by decoding the lumped parameter governing equations formulated in Section 3.2.4 on the basis of the nonlinear parameter relationships measured in Section 3.3. The decoupler dynamics formulated in Section 3.4 is also incorporated into the mathematical model. The values of internal variables at the static equilibrium condition are determined in this section. As a result, the mathematical model may be applied at lower frequencies to the regular hydraulic mount with both a decoupler and an inertia track.

#### 3.5.1 Static Equilibrium Condition

It is apparent at static equilibrium that $p_1 = p_2 = \bar{p}$ and $\bar{V}_1 + \bar{V}_2 = \bar{V}_2$ since $C_1 << C_2$.

It follows from Eqs. (3.16) and (3.25) that

$$-\bar{F} = A_p(5.26\bar{V}_2^{2.5} - 8.9 \times 10^{-5}\bar{V}_2^6 + 1.41 \times 10^{-5}\bar{V}_2^{6.5}) + k_3(\bar{V}_2/A_p) \times 10^{-6} \quad (3.37)$$
where $F$, $A_p$, $k$, and $V_f$ have the units of N, m$^2$, N/m, and cc respectively. Recall that a preload corresponding to the engine weight is applied as a compressive or negative force; therefore, $-\ddot{x} = \frac{V_f}{A_p}$. Equation (3.37) may be solved for $V_f$ for a given $F$. It follows that $\ddot{x}$ is obtained from Eq.(3.25). Figure 3.18(a) shows $p = p(-F)$ and $V_f = V_f(-F)$ curves. Now $V_f$ is obtained from Eq.(3.24) and $-\ddot{x} = (V_f + V_f)/A_p$ in consistent units. Figure 3.18(b) depicts $V_f = V_f(-F)$ and $\ddot{x} = \ddot{x}(-F)$ curves. Initial conditions in the mathematical model are given as $V_i$, $V_f$ and $\ddot{x}$ for an applied $F$. Table 3.2 gives the partial list of their numerical values in the vicinity of $F = -1200$ N.

Table 3.2 Partial list of static equilibrium conditions.

<table>
<thead>
<tr>
<th>$F$ (N)</th>
<th>$p$ (KPa)</th>
<th>$V_f$ (cc)</th>
<th>$\ddot{x}$ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-800</td>
<td>108.4</td>
<td>0.6133</td>
<td>19.206</td>
</tr>
<tr>
<td>-1000</td>
<td>112.1</td>
<td>0.7986</td>
<td>23.761</td>
</tr>
<tr>
<td>-1200</td>
<td>116.4</td>
<td>0.9920</td>
<td>28.251</td>
</tr>
<tr>
<td>-1400</td>
<td>122.6</td>
<td>1.2448</td>
<td>32.480</td>
</tr>
</tbody>
</table>

3.5.2 Inertia Track Dynamics

The momentum equations for two parallel paths $i_1$ and $i_2$ are

\[ p_2(t) - p_1(t) = I_{i_1} \dot{q}_{i_1}(t) + \Delta p_{f_1}(t)q_{i_1}(t)/|q_{i_1}(t)| \]  
(3.38)

\[ p_2(t) - p_1(t) = I_{i_2} \dot{q}_{i_2}(t) + \Delta p_{f_2}(t)q_{i_2}(t)/|q_{i_2}(t)| \]  
(3.39)

The fluid inertances $I_{i_1}$ and $I_{i_2}$ of $i_1$ and $i_2$ are defined as below.

\[ I_{i_1} = \rho g I_{i_1}/A_{i_1} \]  
I
\[ I_{i_2} = \rho g I_{i_2}/A_{i_2} \]  
(3.40a,b)

Note that $\Delta p_f$(Pa)-$q_f$(m$^3$/s) relationships for codes $A$ and $N$ can be derived from Table 3.1.
Figure 3.18  Static equilibrium conditions as a function of preload.
3.5.3 Decoupler Dynamics

By assuming turbulent orifice flow for \( q_d \), \( \Delta p_{fd} \) is given as follows:

\[
\Delta p_{fd}(t) = \left( \frac{1}{C_{de} A_{de}} \right)^\frac{z}{2} \left( \frac{\rho_z}{q_d^2(t)} \right)
\] (3.41)

where \( C_{de} \) is the discharge coefficient and \( A_{de} \) is the effective decoupler area representing the switching mechanism.

\[ A_{de} = A_d \] , \hspace{1cm} \text{for the decoupled state} \hspace{1cm} (3.42)

\[ A_{de} = 0 \] , \hspace{1cm} \text{for the coupled state} \hspace{1cm} (3.43)

The momentum equation for \#d is

\[
p_2(t)-p_1(t)=l_d \dot{q}_d(t)+\Delta p_{fd}(t) q_d(t)/|q_d(t)|
\] (3.44)

The fluidic inerance \( l_d \) of \#d is defined as below.

\[
l\prime_d=p_g l_d A_d
\] (3.45)

where \( l\prime_d \) is the length or height of \#d. The fluid first-order dynamics of \#d as given by Eq.(3.44) is nearly undamped during the decoupled state since \( \Delta p_{fd} \) generated by Eqs.(3.41) and (3.42) is very small. These undamped (or switching) decoupler dynamics cause the pressure ripples in the decoupled state as depicted in Fig.3.17(b).

There are three distinct flow paths between the lower chamber (\#2) and upper chamber (\#1): \( q_{i1}, q_{i2} \) and \( q_d \). However, note that, in the decoupled state, \( q_{i1}=q_{i2}=0 \) since \( \Delta p_{fd} \) exhibited by Eqs.(3.41) and (3.42) is negligible in comparison with \( \Delta p_{f1} \) or \( \Delta p_{f2} \) for a given \( q_i \). On the other hand, in the coupled state, \( q_d=0 \) since \( \Delta p_{fd} \) given by Eqs.(3.41) and (3.43) is much greater than \( \Delta p_{f1} \) or \( \Delta p_{f2} \).

3.5.4 Volumetric Continuity of Fluid Chamber
Defining the total fluid volume transferred between chambers #2 and #1 as $V$,

$$V(t) = \int_0^t [q_{i1}(t) + q_{i2}(t) + q_a(t)] dt$$  
(3.46)

The volumetric continuity for chamber #2 is expressed by its volume increment $V_2$ from the condition that $p_2 = p_{atm}$.

$$V_2(t) = \bar{V}_2 - V(t)$$  
(3.47)

Now, in consistent units, $p_2$ is obtained from the measured $p_2 - V_2$ relationship as given in Eq.(3.25). As a result,

$$p_2(t) = p_2[V_2(t)]$$  
(3.48)

The volumetric continuity for chamber #1 is represented by its volume increment $V_1$ from the condition that $p_1 = p_{atm}$.

$$V_1(t) = \bar{V}_1 + V(t) - A_p x(t)$$  
(3.49)

When $V_1 > 0$, the upper rubber is compressed and $p_1$ is obtained in consistent units from the measured $p_1 - V_1$ relationship as given in Eq.(3.24). As a result,

$$p_1(t) = p_1[V_1(t)], \text{ for } V_1(t) > 0$$  
(3.50)

On the other hand, when $V_1 < 0$, it is assumed that $\bar{V}_{ar}$ is expanded by an amount equal to $|V_1|$ and $p_1$ is obtained from the isothermal process in view of the steady-state. It follows that

$$p_1(t) = p_{atm} \bar{V}_{ar}/(|\bar{V}_{ar} + |V_1(t)||), \text{ for } V_1(t) < 0$$  
(3.51)

Note that the $p_1 - V_1$ relationship in the vacuum state as expressed by Eq.(3.51) is depicted in Fig.3.7(a).

### 3.5.5 Frequency Response
Since $p_1$ may experience vacuum pressures for some duration in a cycle in conjunction with the upward $x$, $F_T$ is asymmetric with respect to the time axis, i.e. its positive and negative peak values are not identical. Furthermore, the response $F_T(t)$ is nonsinusoidal due to nonlinear system parameters, although the excitation is purely sinusoidal; $x(t) = X \sin \omega t$. In the steady state, it may be expressed as shown below by employing a Fourier series expansion,

$$F_T(t) = F_{T0} + \cdots + \hat{F}_{T1} \sin(\omega t + \phi_1) + \hat{F}_{T2} \sin(2\omega t + \phi_2) + \hat{F}_{T3} \sin(3\omega t + \phi_3) + \cdots$$

(3.52)

where $F_{T0}$ is the mean value, $\hat{F}_{T1}$, $\hat{F}_{T2}$, and $\hat{F}_{T3}$ are the amplitudes of each harmonic, and $\phi_1$, $\phi_2$ and $\phi_3$ are the phase leads of each harmonic with reference to $x$. Note that the lower harmonics are totally negligible, and the nonzero $F_{T0}$ exists for the nonlinear hydraulic mount.

Now, $F_{T0}$ is given as,

$$F_{T0} = \frac{1}{nT_p} \int_0^{nT_p} F_T(t) dt$$

(3.53)

where $n$ = number of cycles of $x$ with period $T_p = 2\pi/\omega$. The Fourier filter algorithm (Doebelin, 1980) is employed in order to extract $\hat{F}_m$ and $\phi_m$ ($m=1,2,\cdots$) from the nonsinusoidal steady-state response $F_T(t)$ with reference to $x(t)$. The in-phase or coincident component $P_m(\omega)$ of $F_T$ is given as, where $m=1,2,\cdots$,

$$\frac{1}{nT_p} \int_0^{nT_p} \sin m\omega t \cdot F_T(t) dt = \frac{1}{nT_p} \int_0^{nT_p} \sin m\omega t \cdot \hat{F}_{T1} \sin(m\omega t + \phi_m) dt$$

$$= \hat{F}_m \cos \phi_m = P_m(\omega)$$

(3.54)

The quadrature component $Q_m(\omega)$ is given similarly as,
\[ \frac{1}{nT_p} \int_0^{nT_p} \cos m \omega t F_T(t) dt = \frac{1}{nT_p} \int_0^{nT_p} \cos m \omega t \hat{F}_T \sin(m \omega t + \phi_m) dt \]

\[ \frac{\hat{F}_m \sin \phi_m}{2} = Q_m(\omega) \]  

(3.55)

It follows that

\[ \hat{F}_m(\omega) = 2 \sqrt{P^2_m(\omega) + Q^2_m(\omega)} \]  

(3.56)

\[ \phi_m(\omega) = \tan^{-1}[Q_m(\omega)/P_m(\omega)] \]  

(3.57)

In a similar way as described in Section 2.3, we may define the cross-point dynamic stiffness \( K^{*}(\omega,X) \) in terms of the fundamental harmonic \((m=1)\) as below, which is both frequency and displacement-amplitude dependent but at a uniform temperature.

\[ K^{*}(\omega,X) = K(\omega,X) e^{i \phi_K(\omega,X)} \]  

(3.58)

where \( K \) and \( \phi_K \) are the dynamic stiffness modulus and loss angle which are specifically given as follows:

\[ K(\omega,X) = \hat{F}_T/X; \quad \phi_K(\omega,X) = \phi_1 \]  

(3.59a,b)

In order to generate the frequency domain characteristics of the hydraulic mount, \( K(\omega,X) \) and \( \phi_K(\omega,X) \) are calculated at each \( \omega \) on the basis of Eq.(3.59).

We now define the frequency responses of the internal variables of concern such as \( p_1, q_{i_1} \) and \( q_{i_2} \). Note that the steady-state responses of the internal variables are also nonsinusoidal by the same reasons as for \( F_T(t) \). First, for the frequency response of \( p_1 \), we may not have to employ the Fourier filter algorithm. Maximum and minimum pressure spectra \( p_{1 \text{max}}(\omega) \) and \( p_{1 \text{min}}(\omega) \) may be more meaningful to demonstrate its frequency domain characteristics rather than the amplitude spectra with regard to its fundamental harmonic. Second, for the frequency response of \( q_{i_1} \) and \( q_{i_2} \), we may well employ the Fourier filter algorithm in a similar way as for \( F_T \).
Suppose that the fundamental harmonics are dominant over their higher harmonics.

\[ q_{i1}(t) = Q_{i1}\sin(\omega t + \phi_{q11}) \]  
\[ q_{i2}(t) = Q_{i2}\sin(\omega t + \phi_{q12}) \]

where \( Q_{i1} \) and \( Q_{i2} \) are the amplitudes of each fundamental harmonic, and \( \phi_{q11} \) and \( \phi_{q12} \) are their phase leads with reference to the excitation velocity \( \dot{x} \).

We may define their displacement amplitude-dependent frequency response functions as follows:

\[ Q^*_i(\omega, X) = Q_{i1}(\omega, X)e^{j\phi_{q11}(\omega, X)} \]  
\[ Q^*_i(\omega, X) = Q_{i2}(\omega, X)e^{j\phi_{q12}(\omega, X)} \]

### 3.5.6 Simulation Program

A direct time domain integration software (ACSL, 1991) is employed to efficiently implement the decoupler switching mechanism described in Section 3.4 and to numerically solve all of the governing equations derived in Section 3.5. Appendix E presents the specific algorithm for implementing the decoupler switching dynamics with ACSL (Advanced Continuous Simulation Language) codes. Program 1 in Appendix F lists the ACSL program source codes for the time domain simulation. Program 2 lists the ACSL codes which simulate the frequency responses by implementing the Fourier filter algorithm and finding the minimum and maximum pressures at each \( \omega \).

### 3.6 Verification of Mathematical Model

In this section, the nonlinear mathematical model is verified in both the time and frequency domains by comparing the predicted responses with the
measured signals or data for various excitation amplitudes and frequencies. The steady-state time domain signals were measured in Biomedical Material Testing Laboratory of Davis Medical Research Center at The Ohio State University (site 1), whereas, as mentioned in Chapter II, the frequency domain data were obtained from the material testing laboratory at Teledyne Monarch Rubber Co. in Hartville, Ohio (site 2). The electrohydraulic material testing system of site 1 is MTS 853, whereas the testing system of site 2 is Carl Schenck Hydropuls-System. Note that the latter system is of higher capacity than the former.

In particular, the steady-state time domain responses such as $F_T(t)$ and $P_1(t)$ are verified only for the inertia track mount. The time domain signals for the regular mount could not been recorded due to our limited accessibility to the testing facilities at site 1. However, the fluid dynamic process may be clarified better with the inertia track mount rather than with the regular mount. Highly nonlinear decoupler dynamics could divert the model validation efforts. On the other hand, the frequency domain data such as $K^*(\omega,\lambda)$ measured at site 2 are available only for the regular mount. High amplitude excitation, i.e. $X=1.0$ mm, is impossible to get beyond 15 Hz for the testing system at site 1 due to its limited capacity. Complete lists of the instrumentation and equipment which were employed at sites 1 and 2 are given in Appendices A.4 and A.5 respectively.

3.6.1 Experimental Hardware of Site 1
The experimental facilities employed at site 1 are described in what follows. They are similar to Fig.2.7, and its dynamic testing procedure was presented in Section 2.3.1. In fact, the static load cell does not exist for the
testing system at site 1 but it does at site 2. In common with Chapter II, \( F = -1200 \) N unless specified otherwise.

Figure 3.19 shows the take-apart hydraulic mount and instrumentation detached from the testing system. The dynamic force transducer is an integrated circuit piezoelectric (ICP) type and measures \( F_1(t) \). It has the capability of measuring extremely low frequency signals; discharge time constant is 2000 sec and low frequency limit (-5%) is 0.0003 Hz (PCB, 1984). The dynamic pressure transducer is also an ICP type. It was installed to measure \( p_1(t) \) during high-frequency testing beyond 50 Hz. Since the mathematical model is concerned with lower frequencies, the dynamic pressure transducer is actually not used in this chapter. The power unit is a combination of constant-current power supply and signal amplifier for the ICP transducers. Its function is to power the ICP transducer electronics and to amplify the signal. The strain-gage type absolute pressure transducer measures \( p_1(t) \). It is installed at the side wall of upper rubber such that its diaphragm is flush with the wall. Its sensing diameter is 6.35 mm; compare it with the upper chamber height which is around 8 mm under \( F = -1200 \) N. Therefore this pressure transducer essentially measures an averaged pressure within the upper chamber, which is compatible with the premise of the lumped-parameter model.

If \( p_1 \) does not have to be measured, the production hydraulic mount is directly employed in place of the take-apart mount as shown in Fig.3.20. A digital thermistor is used to measure \( T_g \) inside the mount; the highest \( T_g \) was found to be 40°C during dynamic testing. In addition, \( x(t) \) is measured through an LVDT which is installed inside the hydraulic actuator of the testing system.
Figure 3.19 Take-apart hydraulic mount and instrumentation for component testing.

Figure 3.20 Production hydraulic mount for component testing.
All signals are acquired by and stored in a spectrum analyzer. These are then transferred to VAX 8500 via a 386SX-PC in order to compare the experimental data with analytical results. The Standard Data Format utilities program (Hewlett-Packard, 1991) is employed as a communication protocol between the spectrum analyzer and the PC. Details of data handling are given in Appendix B.

3.6.2 Time Domain Results

The mathematical model is now validated in the time domain with regard to the inertia track mount whose geometric configuration is code N and 2.0 reducer. Along with the transmitted force time histories $F_T(t)$, $F_T-x$ Lissajous diagrams are plotted to facilitate the model validation. This diagram was also used by Vliet and Sankar (1983) for the nonlinear analysis of shock absorbers. Its enclosed area is correlated to the magnitude of $\phi_K$, i.e. damping, and its shape indicates the extent of nonlinearity. Note that $\phi_K=90^\circ$ for a linear viscous damper; a circle or ellipse are plotted depending on $F_T$ and $x$ axis scales. A steady-state response is reached within five cycles in the ACSL simulation. This is because excitation is given by the displacement $x$ with regard to the component testing conditions.

In particular, the measured signal and predicted response of $F_T(t)$ may not be directly compared each other. The measured $F_{T0}$ is basically zero at the steady state since the dynamic force transducer can not measure the nonzero mean value. However, in practice, $F_{T0}$ is not necessarily zero for the nonlinear hydraulic mount as will be described in Fig.3.30. We should subtract the mean component from the predicted $F_T$ in order to compare it
with the measured $F_T$, or a strain-gage type force transducer should have been employed to measure $F_T(t)$.

Now, note that the predicted peak $F_{T0} = -10$ and -45 N at 15 Hz when excited with $X = 0.5$ and 1.0 mm respectively. Since the magnitude of $F_{T0}$ is insignificant in comparison with the amplitude of $F_T$ itself as will be shown shortly, the predicted $F_T(t)$ are directly compared to the measured signals in what follows.

**Effect of Excitation Frequency** Figure 3.21 compares the measured and predicted $F_T(t)$, where the excitation frequency $f$ is increased from 4 Hz to 20 Hz with $X$ being maintained at 0.5 mm. The sinusoidal excitation stroke time history $x(t)$ of the hydraulic actuator is plotted along with $F_T(t)$ in order to show their phase relationship and to assess the extent of nonsinusoidal and asymmetric behavior of $F_T$. In addition, recall the sign convention employed in the mathematical model; the upward $x$ and the corresponding $F_T$ are taken positive as illustrated in Fig.3.1.

Given the lumped-parameter model and other approximations of this theory, predictions match measurements pretty decently up to 10 Hz for both $F_T(t)$ and the $F_T-x$ plots. At 4 Hz, $F_T$ is nearly in-phase with $x$ since the inertia track dynamics are not fully excited yet. As $f$ increases toward 10 Hz, the effect of inertia track dynamics is more noticeable. The phase lead of $F_T$ with respect to $x$ is increased as observed in the time domain responses and the enclosed area of the $F_T-x$ diagram becomes large. These indicate that the fluid damping becomes more evident, and $\phi_K$ gets higher as a result. The frequency domain response $\phi_K(\omega)$ are presented in the following section. In particular, the shape of $F_T$ looks more nonsinusoidal
as \( f \) increases. For instance, it looks triangular at 10 Hz. This apparently results from the nonlinear system parameters such as \( C_1, C_2 \) and \( R_l \).

Now, at 15 Hz, prediction shows some discrepancy from measurement. This signifies that unmodeled dynamics come into effect. The discrepancy becomes more evident at 20 Hz particularly for the \( F_T-x \) diagram. The model does not predict a large \( \phi_k \) as measured in the experiment. This may be related with the simplifying assumptions made for the low frequency model, particularly the exclusion of \( C_{12}(t) \) and \( I_1(t) \). It is explained further in what follows.

**Effect of Excitation Amplitude**  Figure 3.22 examines \( F_T(t) \) at 15 Hz as \( X \) is increased from 0.5 mm to 1.0 mm in a step of 0.1 mm. The enclosed area of the measured \( F_T-x \) diagram increases monotonically with \( X \). However the theory does not predict it well. This discrepancy may be explained as follows. As will be presented in Figs.3.31(b) and (c), the natural frequency \( f_n \) related to the inertia track dynamics is about 13 Hz for code \( N \). As \( f \) is increased beyond \( f_n \), the phase difference between \( q_{i1} \) or \( q_{i2} \) and \( \dot{x} \) approaches to 180°. Therefore \( I_1 \) comes into effect now. During the upward \( x \), some gas may be generated in the upper chamber glycol fluid just above the internal subassembly since \( q_{i1} \) and \( q_{i2} \) flow downward. This gas phase is associated with \( C_{12} \) of Fig.3.1(b). Subsequently, when \( x \) moves down from the positive peak toward the zero (static equilibrium) position, this gas phase may revert to the liquid state. The volume of the generated gas is a function of \( t \) as well as \( X \), as evident from the measured \( F_T-x \) diagram. Since \( C_{12} \) is highly nonlinear and time-varying, it is very difficult to model this phenomenon precisely in the context of the lumped-parameter
Figure 3.21 Effect of excitation frequency on transmitted force; inertia track mount.
Figure 3.21  (continued)

\[ F_T (N) \]

- - - Predicted $F_T$, --- Measured $F_T$
analysis. Note that the main effect of the unmodeled dynamics associated with $I_1(t)$ and $C_{12}(t)$ is to produce a larger $\phi_k$ than the prediction. The gas generation and gas-liquid phase transformation might have been verified by installing a strain-gage type flat pressure transducer on the internal subassembly and measuring $p_{12}$. However the verification was not tried in this study due to time constraints.

On the other hand, the predicted and measured $F_T \cdot \dot{x}$ diagrams exhibit the trend opposite to the $F_T \cdot x$ diagrams. The enclosed area of the predicted $F_T \cdot \dot{x}$ diagram rather increases monotonically with $X$ now.

Figure 3.23 compares the corresponding $F_T(t)$ and $F_T \cdot x$ diagram for each $X$. We observe that $F_T$ becomes more nonsinusoidal and asymmetric as $X$ is increased in both the theory and experiment. For example, for $X=1.0$ mm in the experiment, the positive peak is around 800 N while the negative peak is 1100 N. This may be due mainly to different $C_1$ values during upward and downward $x$; recall the discussion in Section 3.3.1.

**Response Beyond 20 Hz** Figure 3.24 compares $F_T(t)$ at 30 Hz with $X=0.3$ and 0.4 mm; the responses at 20 Hz with $X=0.55$ mm are plotted as a reference. We observe that prediction is considerably different from measurement at 30 Hz due to unmodeled dynamics. Both the phase relationship and the peak values of $F_T$ are significantly different. Let us recall that we did not model two additional dynamic properties of the inertia track: the frequency effect on $R_1$ and the line dynamics phenomenon. Now, we may well raise a question on how much it is feasible for the two unmodeled inertia track dynamics to make contribution to the overall response $F_T$ at 30 Hz. First, as will be discussed in Fig.3.31, $q_{i1}$ and $q_{i2}$ become out of phase as
Figure 3.22 Effect of excitation amplitude on transmitted force at 15 Hz (inertia track mount). (a) Theory, (b) Experiment.
Figure 3.23 Effect of excitation amplitude on transmitted force; inertia track mount.
$f$ is increased. Therefore, the frequency effect on $R_i$ may not be significant. Second, as will be shown in the measured $p_1$ signals of Fig. 3.25(a), the line dynamics effect does not seem to be important. This is because the oscillating inertia track flow becomes more highly damped as $X$ is increased, i.e. $\Delta p_f$ is approximately proportional to the square of the turbulent long-orifice flow $q_i$. As a result, the unmodeled dynamics may largely be related with $l_1$ and $C_{12}$ at 30 Hz. In practice, beyond 20 Hz during the experiment, we can hear the noisy sound of the oscillating upper chamber fluid inertia.

**Upper Chamber Pressures** Figure 3.25(a) depicts the measured $p_1(t)$ at 10 Hz as $X$ is increased from 0.1 mm to 1.0 mm. We observe that the dynamic pressure amplitude becomes higher with $X$ increasing. The measured signals demonstrate that vacuum is reached in the upper chamber; the atmospheric pressure level is indicated by a straight line in each plot. In addition, the measured $p_1$ at 20 Hz with $X=0.5$ mm is also illustrated. Unfortunately, the predicted and measured $p_1(t)$ can not be compared in the same plot, not like the preceding plots for $F_T(t)$. When $p_1$ signal was acquired with the spectrum analyzer, its trigger setting was not based on $x$ but on $F_T$. Since the predicted and measured $F_T$ are not identical in fact, a direct validation of $p_1(t)$ is not feasible. We do not know the phase relationship between the measured and predicted $p_1$.

Now, Figure 3.25(b) shows the predicted $p_1(t)$ and $p_1-x$ diagram by itself at 10 and 20 Hz with $X=1.0$ mm and 0.5 mm respectively. Here $x$ is plotted along with $p_1$ in order to show their phase relationship and to assess the response characteristics of $p_1$. At 10 Hz, $p_1$ exhibits a phase lead of about
Figure 3.24  Transmitted force responses beyond 20 Hz; inertia track mount.
90° with reference to x. Note that the negative or downward x makes p₁ be high. In particular, the p₁-x curve is relatively flat during the upward x. Recalling that the theory matches the experiment reasonably with regard to \( F_T \) at 10 Hz as described in Fig.3.21, this predicted \( p_1(t) \) and \( p_1-x \) diagram may be quite similar to the measured data. On the other hand, at 20 Hz, the phase lead of \( p_1 \) approaches to 0°. By recalling that our theory does not match the experiment well with regard to \( F_T(t) \) at this higher frequency (particularly in their phase relationship), this \( p_1-x \) curve may not be a good prediction of the measurement.

We now compare the minimum and maximum values of \( p_1 \) in Table 3.3. It is observed that the measured minimum pressures are higher than the predicted values. The reason may be explained as follows. When installing the pressure transducer to the side wall of the top element, the inside rubber was detached from its outer metal. Since the gap was not repaired well unfortunately, some air was initially entrapped into it even before the take-apart mount is assembled in the liquid bath. As a result, the entrapped air volume in the upper chamber is larger than the amount assumed in our theory; recall that \( \bar{V}_{air} = 0.025 \text{ cc} \) as described in Section 3.3.1. In practice, the installation of the flush-diaphragm pressure transducer on the top element requires special adaptor and sealant as well as very careful attention, since its outer metal is very thin. Note that, during vibration testing which is presented in Chapter IV, a lower \( (p_1)_{min} \) was observed, say 53.0 KPa (7.7 psia). The gap was repaired better in that experiment.

Even the predicted \( (p_1)_{min} \) is still higher than the vapor pressure of glycol fluid. As a reference, the vapor pressure of \( \text{H}_2\text{O} \) is 15 KPa (2.2 psia) at 54°C. Therefore, our assumption that the negative \( C_f(V_1,p_1) \) is strictly
Figure 3.25  Upper chamber pressure responses of inertia track mount.  
(a) Experiment, (b) Theory.
associated with the expansion of the initially dissolved air is reasonable. In addition, recall that the pressure transducer basically measures the averaged upper chamber pressure as described in Section 3.6.1. In fact, $p_1$ may not be uniform inside the upper chamber at the frequencies beyond $f_n$ as discussed in Fig.3.22.

Table 3.3  Predicted and measured upper chamber pressures of inertia track mount.

<table>
<thead>
<tr>
<th>Excitation</th>
<th>Pressure</th>
<th>Theory</th>
<th>Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$p_1$</td>
<td>KPa (psia)</td>
<td>KPa (psia)</td>
</tr>
<tr>
<td>10 Hz±1.0 mm</td>
<td>$(p_1)_{\text{min}}$</td>
<td>44.9 (6.5)</td>
<td>73.6 (10.7)</td>
</tr>
<tr>
<td></td>
<td>$(p_1)_{\text{max}}$</td>
<td>232.0 (33.7)</td>
<td>226.3 (32.8)</td>
</tr>
<tr>
<td>20 Hz±0.5 mm</td>
<td>$(p_1)_{\text{min}}$</td>
<td>46.4 (6.7)</td>
<td>65.6 (9.5)</td>
</tr>
<tr>
<td></td>
<td>$(p_1)_{\text{max}}$</td>
<td>220.7 (32.0)</td>
<td>260.8 (37.8)</td>
</tr>
</tbody>
</table>

Other Internal Variables  The dynamic responses of $p_2$, $q_{11}$ and $q_{12}$ are not measured. Note that $p_2$ could have been measured by installing a strain-gage type flat pressure transducer just under the internal subassembly; however, it was not tried in this study due to time constraints. On the other hand, $q_{11}$ and $q_{12}$ are rather difficult to measure, since they are not the local flow velocities but volume flow rates over the inertia track cross-sectional area. Therefore, the prediction of $p_2$, $q_{11}$ and $q_{12}$ is one of the features of this lumped-parameter analysis.

Figure 3.26 depicts the effect of excitation frequency on the predicted responses, with $X$ maintained at 0.5 mm. At 4 Hz, $p_2$ is almost in-phase
Figure 3.26  Predicted responses of internal variables; inertia track mount.
with \( x \), and \( q_{i1} \) and \( q_{i2} \) are not far from being in-phase with \( x \). Recall the sign convention as given in Fig.3.5; the negative or downward \( x \) makes \( p_2 \) be high. At 20 Hz, we observe 90° phase lag of \( p_2 \) with respect to \( x \). The phase lags of \( q_{i1} \) and \( q_{i2} \) with reference to \( x \) are around 140° and 90° respectively. Comparing with \( p_1 \) shown in Fig.3.25, the fluctuation amplitude of \( p_2 \) is very small due to the highly compliant \( C_2 \).

We now show the Reynolds number for the inertia track flow so as to examine how much turbulent it is. Figure 3.27 depicts \( Re_{h1} \) and \( Re_{h2} \) at 15 Hz, which are calculated on the basis of the predicted \( q_{i1} \) and \( q_{i2} \) for \( X=1.0 \) mm. These big numbers may indicate that the oscillating inertia track flows are highly turbulent; refer to Section 3.3.3.

![Graph showing \( Re_{h1} \) and \( Re_{h2} \) for \( q_{i1} \) and \( q_{i2} \).](image)

**Figure 3.27** Predicted Reynolds number at 15 Hz, where \( X=1.0 \) mm; inertia track mount.

### 3.6.3 Frequency Domain Results

We defined the dynamic stiffness spectra in Section 3.5.5. In addition, recall the different definitions of frequency responses for \( p_1 \), \( q_{i1} \) and \( q_{i2} \). Now we verify the mathematical model in the frequency domain.
**Dynamic Stiffness Spectra** Figure 3.28 compares the measured and predicted dynamic stiffness spectra of a regular mount, where $X=1.0$ mm. In addition, the predicted $K^*$ of the inertia track mount is plotted as a reference. The predicted time domain responses of the regular mount are presented in Section 3.7.1. As expected, the predicted $K(\omega)$ of the inertia track mount is higher than the measured $K(\omega)$ of the regular mount. With regard to the regular mount, the predicted $K$ and $\phi_K$ are lower than the measured spectra. This implies that the decoupler dynamics are highly nonlinear and more complicated than the lumped-parameter model described in Section 3.4. Table 3.4 lists $\hat{f}_s$ and $\hat{\phi}_K$ of the three spectra. The measured values are in between the predicted values for the inertia track and the regular mounts. We observe that, as $f$ is increased beyond $\hat{f}_s$, the measured $\phi_K$ becomes even larger than the prediction for the inertia track mount. This may be due to the $C_{12}$ and $I_1$ effects as mentioned earlier. Furthermore, the measured $\phi_K$ levels off to some extent beyond 30 Hz while both predictions go down all the way to 50 Hz.

<table>
<thead>
<tr>
<th>Method</th>
<th>Mount</th>
<th>$\hat{f}_s$ (Hz)</th>
<th>$\hat{\phi}_K$ (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Theory</td>
<td>Inertia track mount</td>
<td>9.0</td>
<td>54.5</td>
</tr>
<tr>
<td>Experiment</td>
<td>Regular mount</td>
<td>10.0</td>
<td>40.6</td>
</tr>
<tr>
<td>Theory</td>
<td>Regular mount</td>
<td>11.0</td>
<td>32.6</td>
</tr>
</tbody>
</table>

Figure 3.29 compares the dynamic stiffness spectra in case that $X=0.1$ mm. The predicted and measured $K^*$ match well with regard to the regular
Figure 3.28  Predicted and measured dynamic stiffness spectra for $X=1.0$ mm.
mount. We observe that the dynamic properties of the regular and inertia track mounts are considerably different from each other for this low amplitude excitation. The inertia track mount yields $K$ and $\phi_K$ similar to those shown in Fig.3.28. In particular, their levels become higher as $X$ is decreased from 1.0 mm to 0.1 mm. On the other hand, the regular mount exhibits fairly low $K$ and $\phi_K$ as $X$ is decreased to 0.1 mm. Note that this $K^*$ is similar to that of the low-damping rubber mount as depicted in Fig.2.8(a). This is because the decoupler is totally open; $X_d = 0.16$ mm from Eq.(2.1). It is clarified from Fig.3.29 that the frequency-variant dynamic properties of the regular hydraulic mount comes from the inertia track dynamics, whereas the amplitude-sensitive dynamic properties arise from the decoupler dynamics.

**Nonsinusoidal Characteristics** We now illustrate how much $F_T(t)$ is nonsinusoidal by comparing $\hat{F}_{r_1}(\omega)$ with the amplitude spectra of the higher harmonics and the mean value. Figure 3.30 shows $F_{T_0}(\omega)$, $\hat{F}_{r_1}(\omega)$, $\hat{F}_{r_2}(\omega)$, and $\hat{F}_{r_3}(\omega)$ of the inertia track mount for $X=1.0$ mm; the harmonic amplitude spectra are extracted by using the Fourier filter algorithm as explained in Section 3.5.5. We observe that a nonzero $F_{T_0}$ exists; however, its magnitude is not significant desirably. This is because the mathematical model is based on the production hydraulic mount which is already designed well. As expected, $\hat{F}_{r_1}(\omega)$ is dominant over the other spectra. Therefore, we justify the reason that the dynamic stiffness spectra is defined in terms of the fundamental harmonic.
Figure 3.29 Predicted and measured dynamic stiffness spectra for $X=0.1\,\text{mm}$.
Figure 3.30 Predicted amplitude spectra of transmitted force harmonics of the inertia track mount (Code N, 2.0 reducer); $x$=1.0 mm.

**Frequency Response of Internal Variables** Figure 3.31 shows the predicted spectral characteristics of $p_1$, $q_{i1}$ and $q_{i2}$ for the inertia track mount, where $x$=1.0 mm. Figure 3.31(a) depicts the spectral variation of $(p_1)_{\text{max}}$ and $(p_1)_{\text{min}}$. We observe the phenomenon of upper chamber pressure buildup. In addition, the shape of $p_{1\text{max}}(\omega)$ is very similar to the shape of the corresponding $K(\omega)$ illustrated in Fig.3.28. Note that $(p_1)_{\text{min}}$ reaches 27.3 KPa (4 psia) at 20 Hz. As mentioned earlier, one of the features of this lumped-parameter analysis is that it can predict the frequency response characteristics of the inertia track volume flow rates. Figure 3.31(b) and (c) show $Q_{i1}'$ and $Q_{i2}'$ respectively. Both $Q_{i1}$ and $Q_{i2}$ reach a maximum of 12.56 lpm at 13 Hz, and thereafter they decrease monotonically. Observe that $\phi_{q_{i1}}=-89^\circ$ and $\phi_{q_{i2}}=-48^\circ$ at 13 Hz; compare these with the predicted $\hat{\phi}_x=54.5^\circ$ at 9 Hz as listed in Table 3.4. In particular, as $f$ increases, both $\phi_{q_{i1}}$ and $\phi_{q_{i2}}$ approaches $-180^\circ$. In other words, $q_i$ and $i$ become out of phase. As
Figure 3.31  Predicted frequency responses of internal variables; inertia track mount, Code N, 2.0 reducer, $x=1.0$ mm.
mentioned earlier, this makes the unmodeled dynamics related to \( l_1 \) and \( C_{12} \) come into effect.

**Effect of Inertia Track Length** It was described in Section 2.5 that \( l_i \) is one of the design parameters for adjusting the hydraulic mount dynamic properties. Figure 3.32 compares the variation of \( K^* \) with \( l_{i1} \), where \( X=1.0 \) mm. The measured spectra of the regular mount indicate that, as \( l_{i1} \) becomes shorter from code \( Q \) to code \( A \), \( \hat{f}_i \) is increased and \( \hat{\phi}_x \) becomes smaller. This trend is also predicted by the inertia track mount model. In common with the reasoning as explained in Fig.3.28, the predictions made by the regular mount model are not so decent.

**Effect of Preload** As mentioned earlier, \( F=-1200 \) N in this study. We now examine which effect the different preloads have on the dynamic properties of the hydraulic mount. Recall that, as mentioned in Section 3.5.1, a different \( F \) basically changes the static equilibrium conditions of the mount or the initial conditions of the theory. Figure 3.33 shows the predicted variations of \( K^* \) with \( F \) for the inertia track mount, where \( X=1.0 \) mm. We observe that both \( K \) and \( \phi_x \) do not change noticeably, although, as given in Table 3.2, the initial conditions change considerably when \( F \) varies from -800 N to -1400 N. This insensitivity of \( K^* \) to \( F \) was also confirmed with experiment.

### 3.7 Decoupler Characteristics

In the preceding section, the time domain responses were exhibited only for the inertia track mount. In this section, we illustrate the predicted steady-
Figure 3.32  Effect of inertia track length on dynamic stiffness spectra.
Figure 3.33  Predicted variations of dynamic stiffness spectra with preload for $X=1.0$ mm; inertia track mount, Code N, 2.0 reducer.
state time histories of the regular mount including the decoupler. In addition, the effect of $\Delta_d$ variation on the mount dynamic properties is presented in the frequency domain.

### 3.7.1 Time Domain Response

The predicted responses, such as $F_T(t)$, $p_1(t)$, $p_1(t)$, $q_{i1}(t)$, $q_{i2}(t)$ and $q_d(t)$, of the mount with code $N$, 2.0 reducer and $\Delta_d=0.7$ mm are presented in what follows for a few combinations of $X$ and $\omega$. Figure 3.34 shows the various responses at 10 Hz, where $X=0.1$ mm. We observe from the plots of $q_{i1}$ and $q_{i2}$ that no fluid flows through the inertia track. Therefore, the decoupler is entirely open over the whole period as mentioned in Fig.3.29. Note that $F_T$ is nearly in phase with $x$. Its $F_T-x$ diagram is similar to that exhibited by a typical low-damping rubber element. The $p_1-x$ diagram is nearly a straight line, which denotes that $p_1$ is almost in phase with $x$. The practical meaning of these dynamic responses is that the engine is virtually supported only by the top element, when the relative engine displacement amplitude with respect to the chassis is as low as 0.1 mm. Note that this property results from the very compliant lower rubber bellow.

Figure 3.35 illustrates the response at 10 Hz, where $X=1.0$ mm. We now observe inertia track flows during the coupled state. Note that $q_d$ exists only during the decoupled state. The plot of $p_1$ shows the decoupler dynamics stages 3-6 occurring in the steady state as described in Section 3.4.2. In particular, tiny pressure ripples are exhibited during the decoupled state. This phenomenon is due to the undamped decoupler dynamics as explained in Section 3.5.3, and it is discussed further in Chapter IV. Note the response characteristics of $F_T$ during the coupled and
Figure 3.34 Predicted steady-state time histories of regular mount (Code N, 2.0 reducer, $\Delta_d=0.7$ mm); $x(t)=0.1\sin20\pi t$ mm.
decoupled states. During the decoupled state, $F_T$ follows the shape of $x$. In the $F_T-x$ and $p_1-x$ diagrams, it is plotted as a relatively straight line in comparison with the coupled state. The practical meaning of these dynamic responses is that the fluid damping comes into effect when the relative engine displacement amplitude is as high as 1.0 mm.

Figure 3.36 shows the response at 15 Hz, where $x=1.0$ mm. It looks similar to Fig.3.35. Note, however, the fluid dynamics effect on various responses as $f$ is increased from 10 to 15 Hz. In particular, the relative duration of the decoupled state becomes shorter.

3.7.2 Decoupler-Gap Effect
We now examine the variations of $K^*$ with $\Delta_d$ in Fig.3.37. As $\Delta_d$ is increased from 1.0 to 3.0 mm, both $K$ and $\phi_K$ decrease. Note that such trends are illustrated by the theory as well as the experiment. The quantitative discrepancy between them is due to the highly nonlinear decoupler dynamics as mentioned in Section 3.6.3. For instance, the effect of the decoupler rubber itself on the decoupler fluid dynamics was not taken into account in our theory. The issues related to the unmodeled decoupler dynamics are discussed further in Chapter IV. The decoupler compliance may be taken zero as in the theory since the 90 durometer rubber is very stiff.

3.8 Discussion
Nonlinear chamber compliances, the inertia track steady-state fluid flow resistance and the effective viscosity of the glycol fluid were measured by employing the test setups specifically designed for this study. By examining
Figure 3.35 Predicted steady-state time histories of regular mount (Code $N$, 2.0 reducer, $\Delta_d=0.7$ mm); $x(t)=\text{1.0sin}2\pi t$ mm.
Figure 3.36 Predicted steady-state time histories of regular mount (Code N, 2.0 reducer, Δ_d=0.7 mm); \( x(t) = 1.0 \sin 30\pi t \) mm.
Figure 3.37  Effect of decoupler gap on dynamic stiffness spectra.
the effect of temperature on the glycol fluid viscosity and further on the inertia track fluid resistance, it was clarified that the temperature variation does not significantly affect the hydraulic mount dynamic properties. The nonlinear lumped-parameter mathematical model of the regular mount was formulated by developing the algorithm for the decoupler switching dynamics. In particular, the Fourier filter algorithm was employed to predict the frequency domain properties of the nonlinear hydraulic mount.

The nonlinear mathematical model was verified by comparing the predicted responses with the component testing results with regard to the transmitted force and upper chamber pressure signals, and dynamic stiffness spectra. The nonlinear model matches the experiment reasonably up to 20 Hz. However, beyond 20 Hz, some discrepancies are observed between the theory and experiment due to the unmodeled dynamics related with the upper chamber fluid inertia and fluid compliance effect associated with the gas-phase generation. Furthermore, the actual decoupler dynamics are much more nonlinear than the proposed switching dynamics. This is due to the unmodeled dynamics effect inside the decoupler control volume.

It was shown from the measured upper chamber pressure signals that the line dynamics effect of the inertia track is not important for the hydraulic mount dynamic responses. Even though the frequency effect on the inertia track flow resistance is believed not to be significant, it needs to be verified with measurement. However, the oscillating turbulent flow resistance is rather difficult to measure with the present instrumentation technology. As a whole, the fluid first-order inertia track dynamics and the
decoupler switching dynamics of the mathematical model reasonably represent the excitation frequency and deflection amplitude dependent dynamic characteristics of the hydraulic mount.
CHAPTER IV
PERFORMANCE EVALUATION OF PASSIVE HYDRAULIC MOUNT IN VEHICLE MODEL

4.1 Introduction
In Chapter II, by employing the quasilinear analysis as an approximate methodology, we examined the dynamic performance of the passive hydraulic mount during steady-state vehicle operation. Now that the nonlinear mathematical model has been developed for the hydraulic mount in Chapter III, we may rigorously evaluate the performance features of the passive mount with regard to a vehicle model in this chapter. The vehicle dynamics are examined with harmonic and impulse excitations. The frequency range of concern is 3 to 20 Hz for the harmonic responses. In addition, laboratory vibration testing is carried out for engine-mounting systems in order to augment the analysis results.

4.2 Mathematical Formulation for Vehicle Model
A vehicle model including a generic nonlinear engine mount and suspension system was illustrated in Fig.2.11. Now, by replacing the nonlinear components with a hydraulic mount and linear suspension system, the vehicle model may look like Fig.4.1. Recall that the suspension system is represented by a Voigt model as mentioned in Section 2.4.2. In addition, \( y \) is the road profile disturbance with reference to the suspension
system. The masses of the top element and canister of the hydraulic mount are incorporated into $m_e$ and $m_s$ respectively since they are ignorable in the context of the vehicle model as described in Section 2.4.2.

![Figure 4.1 Vehicle model with a hydraulic mount and linear suspension system.](image)

Now we formulate the equations of motion in what follows. Applying Newton’s second law to $m_e$, we may obtain

$$F(t)-k_r[x_e'(t)-x_s'(t)]-b_r[x_e'(t)-x_s'(t)]+A_p[p_1(t)-p_{atm}]-m_e g = m_e \ddot{x}_e(t) \quad (4.1)$$

where

$$x_e'(t) = \ddot{x}_e + x_e(t) \quad ; \quad x_s'(t) = \ddot{x}_s + x_s(t) \quad (4.2a,b)$$

The equilibrium condition may be given by,

$$A_p(\bar{p}-p_{atm})-\bar{k}_r(\ddot{x}_e - \ddot{x}_s) = m_e g \quad (4.3)$$

By using Eqs.(4.2) and (4.3), we may rewrite Eq.(4.1) in terms of the dynamic displacements $x_e$ and $x_s$ as follows:

$$F(t)-k_r[x_e(t)-x_s(t)]-b_r[x_e(t)-x_s(t)]+A_p[p_1(t)-\bar{p}] = m_e \ddot{x}_e(t) \quad (4.4)$$

Applying Newton’s second law to $m_s$ in terms of total components, we may obtain
\[-k_r[x'_r(t)-x'_e(t)]-b_r[x'_r(t)-\ddot{x}_e(t)]-k_s[x'_s(t)-y'(t)]-b_s[\ddot{x}_s(t)-\ddot{y}(t)]
\]
\[-A_p[p_1(t)-p_{atm}]=m_s\ddot{x}_s(t) \tag{4.5}\]

where
\[y'(t)=\ddot{y}+y(t) \tag{4.6}\]

The corresponding static equilibrium condition may be given by,
\[-k_s(\ddot{x}_s-\ddot{x}_e)-k_s(\dddot{x}_s-\dddot{y})-A_p(\dddot{p}-p_{atm})=m_s\dddot{y} \tag{4.7}\]

By using Eqs.(4.6) and (4.7), we may rewrite Eq.(4.5) in terms of dynamic displacements as below.
\[-k_r[x_e(t)-x_e(t)]=b_r[x_e(t)-\ddot{x}_e(t)]-k_s[x_s(t)-y(t)]-b_s[\ddot{x}_s(t)-\ddot{y}(t)]
\]
\[-A_p[p_1(t)-\dddot{p}]=m_s\ddot{x}_s(t) \tag{4.8}\]

4.3 Analysis of Vehicle Model

All of the governing equations of the hydraulic mount as given in Eqs.(3.37)-(3.53) are exactly the same in the vehicle model. In particular, recall that \(\dddot{F}=-m_eg\), and \(x\) is equivalent to the relative engine displacement as given by Eq.(2.32). Both the steady-state and transient performances of the hydraulic mount are evaluated and compared with those of the low-damping rubber-metal mount. Unless specified otherwise, the geometric configuration of the hydraulic mount is Code \(N\) and 2.0 reducer. The mechanical properties of the rubber mount were given in Fig.3.2; the effect of deflection amplitude on \(k_r(\omega)\) and \(b_r(\omega)\) is not taken into account. The excitation for the steady-state response is given by the harmonic engine force, whereas the excitation for the transient response is provided by an impulsive or shock load acting on \(m_e\). In this analysis, the road profile is taken to be uniform, \(i.e. y(t)=0\). Vehicle model parameters corresponding to a typical medium size passenger car were given in Section 2.4.3.
4.3.1 Harmonic Excitation

The best way for applying $F(t)$ to the vehicle model is to measure the unbalanced engine force from actual engine-block testing at each engine speed. The actual engine force is certainly not harmonic but periodic. However, in this study, let us assume that $F$ is harmonic and furthermore that its amplitude is uniform over the frequency range 3 to 20 Hz.

$$F(t) = F_0 \sin \omega t$$

(4.9)

The vehicle model under this harmonic excitation may simulate the vehicle dynamics arising during engine idle.

Since the hydraulic mount is a highly nonlinear dynamic device, the vehicle model exhibits non-sinusoidal response as will be shown shortly although the excitation is harmonic. For instance, in the steady state, the relative engine displacement $x$ may be expressed as below by employing a Fourier series expansion,

$$x(t) = X_0 + \cdots + X_1 \sin(\omega t + \phi_{x1}) + X_2 \sin(2\omega t + \phi_{x2}) + X_3 \sin(3\omega t + \phi_{x3}) + \cdots$$

(4.10)

where $X_0$ is the mean value, $X_1$, $X_2$ and $X_3$ are the amplitudes of each harmonic, and $\phi_{x1}$, $\phi_{x2}$ and $\phi_{x3}$ are the phase leads of each harmonic with reference to $F(t)$. Note that the lower harmonics are negligible. Since the fundamental harmonic is dominant over the mean value and the higher harmonics, we may define the relative engine displacement amplitude spectrum $X(\omega)$ as follows:

$$X(\omega) = X_1(\omega)$$

(4.11)

The engine acceleration $\ddot{x}_e$ and the sprung mass acceleration $\ddot{x}_s$ may also be expressed by employing a Fourier series expansion,

$$\ddot{x}_e(t) = \cdots + \ddot{X}_{e1} \sin(\omega t + \phi_{e1}) + \ddot{X}_{e2} \sin(2\omega t + \phi_{e2}) + \ddot{X}_{e3} \sin(3\omega t + \phi_{e3}) + \cdots$$

(4.12)
\[ \ddot{x}_i(t) = \ldots + \ddot{x}_{i1} \sin(\omega t + \phi_{i1}) + \ddot{x}_{i2} \sin(2\omega t + \phi_{i2}) + \ddot{x}_{i3} \sin(3\omega t + \phi_{i3}) + \ldots \]  (4.13)

where \( \ddot{x}_{i1}, \ddot{x}_{i2}, \ddot{x}_{i3}, \ddot{x}_{i4}, \ddot{x}_{i5} \) and \( \ddot{x}_{i6} \) are the amplitudes of each harmonic, and \( \phi_{i1}, \phi_{i2}, \phi_{i3}, \phi_{i4}, \phi_{i5} \) and \( \phi_{i6} \) are the phase leads of each harmonic with reference to \( F(t) \). Note that the lower harmonics are negligible. In common with the definition of \( X(\omega) \), the engine and sprung mass acceleration amplitude spectra \( \ddot{x}_e(\omega) \) and \( \ddot{x}_s(\omega) \) may be defined in terms of the fundamental harmonics as below.

\[ \ddot{x}_e(\omega) = \ddot{x}_{e1}(\omega) ; \quad \ddot{x}_s(\omega) = \ddot{x}_{s1}(\omega) \]  (4.14a,b)

Note that these amplitude spectra are obtained by using the Fourier filter algorithm described in Section 3.5.5. In general, phase spectra are not of practical concern in vehicle dynamics.

The ACSL program source codes for the time domain simulation are listed as Program 3 in Appendix F, whereas the frequency domain simulation codes are given as Program 4. In what follows, we present the simulation results in terms of frequency domain and starting transient responses.

**Frequency Domain Response** Figure 4.2 compares the hydraulic and the low-damping rubber-metal mounts with regard to \( \ddot{x}_e(\omega), \ddot{x}_s(\omega) \) and \( X(\omega) \), where \( F_d = 100 \) N. A high peak resonance occurs at around 9.2 Hz for the rubber mount. The hydraulic mount clearly shows its superior dynamic performance: resonance control and vibration isolation beyond the engine resonant frequency. The resonant level is significantly reduced by the fluid damping dynamics of the inertia track. In particular, the peak \( \ddot{x}_e \), usually taken as one of the indices evaluating the occupant ride quality, is decreased from 3.3 m/s\(^2\) to 0.7 m/s\(^2\). In addition, the peak \( \ddot{x}_e \) is drastically
reduced from 7.5 m/s² to 2.0 m/s². On the other hand, for \( f \geq 15 \) Hz, the decoupler action \((\Delta_d = 0.7 \text{ mm})\) yields low \( \ddot{x}_r \) and \( \dddot{x}_s \), as similar to the response produced by the low-damping rubber mount. It turns out that \( X/X_d = 0.16 \text{ mm for } f \geq 15 \text{ Hz} \); refer to Eq.(2.1). This implies that the decoupler is totally open at the frequencies beyond or equal to 15 Hz.

As mentioned in Section 2.4.3, the problematic aspect of the passive hydraulic mount at lower frequencies is also observed in Fig.4.2. For instance, beyond 11 Hz and until the decoupler is totally open at 15 Hz, the hydraulic mount yields a higher \( \dddot{x}_s \) than the rubber mount. This performance limitation may result from the stiffening phenomenon or upper chamber pressure buildup of the hydraulic mount.

**Non-sinusoidal Characteristics** We now illustrate how much the steady-state responses of \( \dddot{x}_r, \dddot{x}_s, \) and \( x \) are non-sinusoidal by comparing their fundamental harmonic amplitude spectra with the corresponding higher harmonic amplitude spectra. As shown in Fig.4.3, \( \dddot{x}_i(\omega), \dddot{x}_j(\omega) \) and \( X_j(\omega) \) are dominant over their higher harmonic amplitude spectra; \( X_0(\omega) \) is negligible. This justifies the reason that \( \dddot{x}_r(\omega), \dddot{x}_s(\omega), \) and \( X(\omega) \) are defined in terms of the fundamental harmonics as in Eqs.(4.14) and (4.13) respectively.

**Effect of Inertia Track Length** Figure 4.4 compares the frequency response spectra for codes \( A \) and \( N \), where \( F_d = 100 \text{ N} \). The responses for the rubber mount are plotted together as a reference. We observe the effect of \( l_i \) on various spectra. In comparison with code \( A \), code \( N \) produces more discontinuity at the boundary frequency between the coupled
Figure 4.2 Acceleration and relative engine displacement amplitude spectra of vehicle model; $F_a=100$ N.
Figure 4.3 Amplitude spectra of the acceleration and relative engine displacement harmonics for hydraulic mount (Code $N$, $\Delta_d=0.7$ mm); $F_a=100$ N.
and decoupled states. This trend is also exhibited in vibration testing as presented in Section 4.4.2. Furthermore, recall that, as illustrated in Fig.3.32, the regular mount model does not predict the effect of \( l_i \) on the mount dynamic properties as much as the experiment does. Therefore, the practical variation of the response spectra with \( l_i \) may be more noticeable than that shown in Fig.4.4. This is verified in Section 4.4.2.

**Effect of Decoupler Gap** Figure 4.5 shows the frequency response spectra for the hydraulic mounts with \( \Delta_d = 0, 0.7 \) and 1.4 mm, where \( F_a = 100 \) N. Note that \( \Delta_d = 0 \) represents the inertia track mount as depicted in Fig.3.1(a). When \( \Delta_d \) is increased from 0.7 to 1.4 mm, the resonance control characteristic is deteriorated and the decoupler begins to totally open at a lower frequency as expected. As \( \Delta_d \) increases more, the dynamic response may eventually approach that for the rubber mount. In consequence, the larger \( \Delta_d \) produces more discontinuity in the response spectra.

Interestingly, the inertia track mount yields different dynamic response from the regular mount. Below 15 Hz, this mount produces a lower \( \ddot{x} \) than the regular mounts and rubber mount. Recall that the regular mount behaves like a rubber mount during its decoupled state at lower frequencies. In other words, the regular mount is not effective in absorbing engine resonance for \( x < X_d \) due to its inherent passive nature. However, beyond 15 Hz, \( \ddot{x} \) for the inertia track mount gradually increases rather than decreasing as for the regular mount. This may be the reason that the inertia track mount can not be employed as an engine mount by itself; its vibration isolation property is obviously inadequate. Figure 4.6 shows the response spectra for \( F_a = 500 \) N. As may be anticipated, vibration
Figure 4.4 Effect of inertia track length on frequency response spectra; $F_a=100$ N.
Figure 4.5 Effect of decoupler gap on frequency response spectra; \(F_d=100\) N.
levels are higher than in Fig.4.5. However, the general features remain the same.

In summary, the inertia track mount has excellent resonance control characteristic while the low-damping rubber mount exhibits superior vibration isolation property. This observation leads to the design of a new adaptive hydraulic mount system in Chapter V.

**Starting Transient Response**  We now present the time domain responses in terms of starting transients till the steady state is reached. Figure 4.7 shows $\ddot{x}_e(t)$, $\dot{x}_e(t)$, and $x(t)$ for the rubber mount leading to resonance at 9.5 Hz, where $F_o=100$ N. Figure 4.8 illustrates the resonance control characteristic of the hydraulic mount ($\Delta_d=0.7$ mm). In the plot of $p_1(t)$, we observe the various stages of decoupler dynamics as described in Section 3.4.2. In particular, the undamped decoupler dynamics induce tiny pressure ripples during the decoupled states ($q_{i_1}=0$) as explained in Section 3.5.3. It is apparent that the inertia track flow, *i.e.* fluid resistance, such as $q_{i_1}$ in the coupled state damps down the engine resonance. In addition, $\ddot{x}_e$ and $\dot{x}_e$ are considerably non-sinusoidal. The irregular shape of $\ddot{x}_e$ results from the decoupler switching dynamics; this will be described further in Section 4.4.2. Figure 4.9 shows the various responses at 14 Hz. In particular, the shape of $\ddot{x}_e$ is more regular than in Fig.4.8.

Figure 4.10 illustrates the vibration isolation characteristic of the hydraulic mount at 15 Hz. During the initial transient state, $q_{i_1} \neq 0$ since the amplitude of $x$ is large enough to close the decoupler. After the transient state has been damped down, $q_i = 0$ and $p_1 = p_2$ since the decoupler is now totally open. Therefore, the engine is supported virtually by only the top
Figure 4.6 Effect of decoupler gap on frequency response spectra; $F_d=500$ N.
Figure 4.7  Starting transients for rubber mount near resonance; $F(t)=100\sin19\pi t$ N.
Figure 4.8 Starting transients for hydraulic mount (Code N, $\Delta_d=0.7$ mm); $F(t)=100\sin14\pi t$ N.
Figure 4.9 Starting transients for hydraulic mount (Code $N$, $\Delta_d = 0.7$ mm); $F(t) = 100\sin 28\pi t$ N.
Figure 4.10 Starting transients for hydraulic mount
(Code $N$, $\Delta_d = 0.7$ mm); $F(t) = 100 \sin 30\pi t$ N.
element. Observe a low level of $\ddot{x}_s$ in the steady state. Figure 4.11 shows the starting transients at 10 Hz for $\Delta_d = 1.4$ mm. We observe the nonuniform responses during the initial transient state.

### 4.3.2 Impulse Excitation

In general, the system response to an impulsive input is of considerable importance. Specifically, the impulse or shock response of the vehicle model may simulate the operating conditions such as the abrupt acceleration, braking, and when the vehicle encounters a road bump or hole during maneuver. Recall that an ideal impulse has infinite "height", infinitesimal duration, but a finite area. However, no real physical variable can behave in precisely this fashion (Doebelin, 1972). In this study, the ideal impulse is simulated as the initial velocity of $m_c$; $\dot{x}_s(0) = 0.1$ m/s. It follows that the impulsive force area, $I_L = m_c\dot{x}_s(0) = 12.23$ N·s. The top element parameters $k_1$ and $b_z$ are taken by their nominal values at 10 Hz.

#### Impulse Response

Figure 4.12 compares the vehicle model impulse responses or shock absorption properties of the low-damping rubber mount and a regular hydraulic mount ($\Delta_d = 0.7$ mm). Although the hydraulic mount produces higher first peaks of $\ddot{x}_r$ and $\ddot{x}_s$, it quickly damps down their subsequent responses and stabilizes $x$. Therefore, in addition to yielding excellent harmonic responses, the hydraulic mount may improve both ride quality and vehicle durability for impulsive inputs.

#### Effect of Decoupler Gap

Figure 4.13 compares the inertia track mount ($\Delta_d = 0$ mm) and a regular mount ($\Delta_d = 1.4$ mm) with regard to the impulse
Figure 4.11  Starting transients for hydraulic mount
(Code N, $\Delta_d = 1.4$ mm); $F(t) = 100\sin 20\pi t$ N.
Figure 4.12  Impulse response of vehicle model; $\dot{x}_e(0)=0.1$ m/s.
response. We observe that the inertia track mount is more effective in absorbing the shock input. The regular mount is in the decoupled state after the initial transient state, thus $q_{ii} = 0$ and fluid damping does not arise. The oscillating frequency of $\ddot{x}_r$, $\ddot{x}_i$ and $x$ during the decoupled state coincides with the engine resonant frequency (9.2 Hz) which is exhibited in the impulse response of the rubber mount as shown in Fig.4.12. On the other hand, the inertia track mount can function as a shock absorber at any moment since it is always in the coupled state.

4.4 Vibration Testing of Engine-Mounting System

If we want to verify the harmonic response of the vehicle model as presented in the preceding section, we should prepare a force source to simulate the unbalanced engine force given by Eq.(4.9). An electrohydraulic actuator operating under a closed-loop force control may be appropriate to produce a sinusoidal force of the selected amplitude and frequency. However, the time allocated for this study does not allow building this test setup. In addition, we should note that the dynamic properties of the suspension system are not linear as presumed in the simulation; they are another research topic by itself. Therefore, a force source is not prepared in this study. Now we may also realize that there is no absolute necessity to verify the vehicle model simulation results. We have already verified the mount mathematical model in Chapter III, and the other vehicle model parameters are just straightforward.

As a matter of fact, the best way to evaluate the performance characteristics of the hydraulic mount is to carry out in-situ vehicle testing as many engineers have done. However, field testing is not appropriate for
Figure 4.13  Impulse response for inertia track and regular hydraulic mounts; $\dot{x}(0)=0.1$ m/s.
our comprehensive study on the dynamic performance of hydraulic mounts because the excitation sources for the vehicle dynamics are rather complicated. We need laboratory testing that is operator-controllable.

With this background and in order to augment the vehicle model simulation, we now employ an electrodynamic vibration testing system which is readily available in the Vibration Laboratory of the Department of Engineering Mechanics at The Ohio State University. The schematic diagram of the vibration testing setup is illustrated in Fig.4.14. A motion input is applied to the engine-mounting system through the electromagnetic shaker table.

4.4.1 Experimental Hardware

Each component or apparatus is identified in the following pictures. Figure 4.15 shows the overall view of the experimental setup. Figure 4.16 shows the engine-mounting system and the electrodynamic shaker. The hydraulic mount is located between the shaker table and the engine mass. Two 45.36 Kg iron weights are utilized to simulate a portion of the vehicle engine block mass. The iron weights are clamped to the plate which is installed on top of the mount. They are supported by frictionless linear bearings with one at each side such that only the vertical motion is feasible. The bearing shaft is fixed to the supporting shoulder of the shaker. The equivalent $m_e$ including the masses of two linear bearings and the connecting plates is 110 Kg. Harmonic or shock excitation $\ddot{x}_i$ is applied through the shaker table and the dynamic response $\ddot{x}_e$ of $m_e$ is recorded. The signal generator and two charge amplifiers are also shown in the picture.
Figure 4.14  Setup for vibration testing of engine-mounting system.
Figure 4.15  Setup with vibration testing, measurement and data acquisition systems.

Figure 4.16  Engine-mounting system on an electrodynamic shaker.
Figure 4.17 shows a detailed view of the engine mount installation on the shaker table through a mounting fixture, which is solidly fixed to the shaker table by four 3/8-16 threaded bolts. Actually, this picture shows the take-apart hydraulic mount which is employed whenever $p_1(t)$ is to be measured. It is the identical one which was employed in component testing as described in Section 3.6.1. If $p_1$ does not have to be measured, the production hydraulic mount is installed directly on top of the mounting fixture. We see two accelerometers; one is installed on the mounting fixture to measure $\ddot{x}_i(t)$ and the other on the plate below the iron weights to measure $\ddot{x}_e(t)$. Both are installed through their mounting studs. Figure 4.18 shows the strain-gage type pressure transducer installed on the top element outer metal of the take-apart hydraulic mount. We see the clear configuration of the mounting fixture. It is made up of solid-cylinder steel block, whose outside diameter is 6 inch and height is 2 inch.

Figure 4.19 shows the LVDT (Linear Variable Differential Transformer) which measures the relative engine displacement $x$ with respect to the shaker table during shock testing.

$$x(t) = x_0(t) - x_i(t) \quad (4.15)$$

Since it could not be connected directly in between the iron weights and shaker table, a metal strip extension is employed. In particular, this LVDT is not used during harmonic testing; it has been found that the metal strip and the LVDT at its free-end vibrates by themselves in the cantilever-beam mode. Figure 4.20 shows the data acquisition system or two-channel spectrum analyzer along with the strain-gage conditioner for the pressure transducer. The signals displayed on the analyzer screen are $p_1(t)$ in the upper trace and $\ddot{x}_i(t)$ in the lower trace. This data acquisition system is the
Figure 4.17  Installation of take-apart hydraulic mount.

Figure 4.18  Strain-gage type pressure transducer installed on hydraulic mount.
Figure 4.19  LVDT installation.

Figure 4.20  Data acquisition system and strain-gage conditioner.
identical one which was employed in component testing as described in Section 3.6.1.

Now, all the components have been identified. Appendix A.6 lists their pertinent manufacturer's specifications. In what follows, the features of some components are explained in detail and the specifications are referenced where appropriate.

**Electrodynamic Shaker** The vibration testing system is not a closed-loop type but a basic open-loop type. Recall that the electrohydraulic material testing system employed in the component testing of engine mounts is operated under closed-loop control. A command voltage signal is produced by a signal generator with 50 Ω output impedance and fed to the solid-state power amplifier. The electromagnetic shaker can operate in two modes: sine and random-input modes. The sine-input mode is selected for harmonic vibration testing, whereas the random-input mode is chosen for shock testing. During the normal operation, this vibration testing system consumes 80 A current from the 220 VAC power line. The shaker capacity is 4900 N sine or 3225 N random peaks. Its stroke range is ±5.08 cm. The static deflection of the shaker table with the engine mounting system installed is 1.3 cm. The engine mounting system weighing 1079 N should not remain installed unless vibration testing is carried out; otherwise, the flexure element supporting the shaker table may undergo a permanent deflection. One of the inherent characteristics of the electromagnetic shaker is that it is not effective for dc command voltages. Therefore, the static displacement is hardly generated by the shaker table.
Mounting Fixture  When we carry out vibration testing on an object, the mounting fixture is generally required. It should be specifically prepared for each individual experiment and firmly fastened to the shaker table such that it does not appear to be a spring element but just a mass element. In general, an aluminum block is recommended due to its higher ratio of elastic modulus to density rather than the steel block. There was no specific reason to employ the steel block in this experiment.

Linear Guide Bearings  The iron weights should be clamped to be well balanced on the plate such that the bearing blocks travel uniformly at both sides. A few drops of machine oil are quite effective in lubricating the linear ball bearings.

Accelerometers  In piezoelectric accelerometers, the low-frequency response is limited by the piezoelectric characteristic, while the high-frequency response is limited by mechanical resonance (Doebelin, 1990). Therefore, in general, smaller accelerometers have a good high frequency performance, whereas larger ones can yield a better low frequency dynamic performance. Since the frequency range in harmonic testing goes up to only 20 Hz, larger accelerometers (spec A.6.5c) are employed in this experiment. They have high charge sensitivity (98 pC/g), and their high frequency limit is far more than enough even for shock testings in this study. By selecting the 0.2 Hz (10 %) acceleration-mode of the high input-impedance charge amplifier, the effective lower frequency limit may go down to 2 Hz.
LVDT  This LVDT is not equipped with the voltage regulator, and 10 VDC is applied as the input voltage. The original output signal from the LVDT contains $\pm 10$ mV ($=\pm 0.07$ mm) noise. Therefore, the signal of $x(t)$ is acquired through the anti-aliasing filter installed in the spectrum analyzer. Its stroke range is $\pm 5.08$ mm.

In what follows, we examine both the steady-state and transient dynamic performances of the passive hydraulic mount along with those of the low-damping rubber-metal mount. Three hydraulic mounts employed in this experiment have codes A, N and Q respectively, but the same internal configuration of $2.0$ reducer and $\Delta_d=0.7$ mm. As mentioned in Section 2.3.2, the rubber mount is essentially the top element of the hydraulic mount.

4.4.2 Harmonic Excitation

A 20 V peak-to-peak sinusoidal command signal is fed to the power amplifier. The input and output accelerations $\ddot{x}_i$ and $\ddot{x}_o$ are continuously monitored with the spectrum analyzer. As we will see shortly, both $\ddot{x}_i$ and $\ddot{x}_o$ are non-sinusoidal in this harmonic testing. Their fundamental harmonics $\ddot{x}_{i1}$ and $\ddot{x}_{o1}$ may be expressed as below for each testing frequency $\omega$.

$$\ddot{x}_{i1}(t)=\dddot{x}_i \sin \omega t$$  \hspace{1cm} (4.16)

$$\ddot{x}_{o1}(t)=\dddot{x}_o \sin (\omega t+\phi_{xo})$$  \hspace{1cm} (4.17)

where $\dddot{x}_i$ and $\dddot{x}_o$ are the amplitudes of $\ddot{x}_{i1}$ and $\ddot{x}_{o1}$ respectively, and $\phi_{xo}$ is the phase lead of $\ddot{x}_{o1}$ with reference to $\ddot{x}_{i1}$.

As a benchmark for obtaining the frequency response of this nonlinear system, the gain-setting knob on the front panel of the vibration
testing system is adjusted at each $\omega$ such that $\ddot{X}_i$ is maintained at a constant value of 1.7 m/s² (rms value=1.2 m/s²). This may yield a practical range of vibration levels over the frequency range of concern, i.e. the fundamental displacement amplitude is reduced with increasing $\omega$ in proportion to $\omega^2$.

**Frequency Domain Response** Figure 4.21(a) shows the output acceleration amplitude spectra $\ddot{X}_o(\omega)$ for the three hydraulic and rubber mounts. Figure 4.20(b) depicts their phase spectra $\phi_{xo}(\omega)$. All spectra are determined from the spectrum analyzer at each individual $\omega$ in the frequency range 3 to 20 Hz. Note that, in the charge amplifier, the low frequency limit is set to the 0.2 Hz-acceleration mode while the upper frequency limit is set to 1 KHz. In general, the measured harmonic accelerations in the frequency range between one decade above the low frequency limit and one decade below the upper frequency limit are considered to be reliable (Bruel & Kjaer, 1982). Therefore the acceleration measurement results over 3 to 20 Hz may well be credible. An averaging process for the frequency response calculations is not necessary since the engine-mounting system is well balanced.

For the rubber mount, $\ddot{X}_o=10$ m/s² at the resonant frequency of around 7 Hz. In practice, it is difficult to measure since the vibration level is too high; caution should always be exercised not to damage the electrodynamic shaker at this resonance. Note that its $\phi_{xo}$ shifts abruptly from $0^\circ$ to $-180^\circ$. This signifies that the rubber mount is nearly undamped. The hydraulic mounts exhibit resonance control characteristics as similar
to those described in Fig.4.2(b) for the engine force excitation. Their plots of \( \phi_{xo}(\omega) \) indicate that the hydraulic mount is highly damped.

In addition, the variations of \( \ddot{X}_o(\omega) \) and \( \phi_{xo}(\omega) \) with codes A, N and Q confirm that \( l_f \) is a design parameter to tune the engine-mounting system dynamics of a vehicle as explained in Section 2.5. As mentioned in Fig.4.4, these vibration testing results show more significant effect of \( l_f \) on the frequency domain responses than the vehicle model simulation predicts. The problematic aspect of the passive hydraulic mount due to the upper chamber pressure build-up is also observed in the plot of \( \ddot{X}_o(\omega) \).

Figure 4.21(c) depicts the spectra \( \chi(\omega) \) of the peaks of \( x(t) \). As will shortly be shown in the steady-state time histories, \( x(t) \) is not harmonic but nearly sinusoidal. We observe that \( \chi \) decreases with increasing \( f \). The accelerometers are used to measure \( x \) along with the charge amplifiers set to their 1.0 Hz-displacement mode. In other words, by presuming that the acceleration signals \( \ddot{x}_i \) and \( \ddot{x}_o \) are nearly harmonic, they are integrated twice with 1.0 Hz (10%) lower frequency limit to yield \( x_i \) and \( x_o \) (Bruek & Kjaer, 1982). Now \( x \) is obtained by subtracting \( x_i \) from \( x_o \) as given by Eq.(4.15). This function can be carried out by using either the spectrum analyzer or a 4-channel digital oscilloscope as depicted in Fig.4.14. The frequency response attenuation effect of the charge amplifier in its displacement mode is compensated for in order to obtain the "true" displacement. The detailed calibration procedure is presented in Appendix C.

In practice, as will be shown shortly, \( \ddot{x}_o \) is not harmonic and \( \ddot{x}_i \) is especially far from being harmonic. Therefore, the displacement signals \( x_i \) and \( x_o \) provided by the charge amplifier are not true responses. In other
words, $x(\omega)$ depicted in Fig.4.21(c) contains some degree of errors. However, as mentioned earlier, these data are still better than those measured by the LVDT.

Returning to Fig.4.21(a), like the low-damping rubber mount, the hydraulic mounts with codes $N$ and $Q$ exhibit vibration isolation characteristics for $f \geq 16$ Hz whereas the mount with code $A$ provides vibration isolation for $f \geq 17$ Hz. Note in Fig.4.21(c) that $x < 0.24$ mm for $f \geq 16$ Hz with regard to codes $N$ and $Q$. The decoupler may stay decoupled when the vibration isolation properties are exhibited. In particular, for the hydraulic mount, the measured $\ddot{x}_o$ is relatively higher than the predicted $\ddot{x}_r$ with reference to the rubber mount response for each case. For instance at 20 Hz, $\ddot{x}_o$ of code $N$ is 50% higher than that of the rubber mount in Fig.4.21(a), whereas $\ddot{x}_r$ of code $N$ is only 10% higher than that of the rubber mount in Fig.4.22(b). As will be explained later, this may result from the undesirable side effect of the chamber fluid inertia which is not included in our theory.

Note that the inertia track mount is not employed in this vibration testing. It is believed that the mount may produce the similar performance characteristics as exhibited in the vehicle model simulation: superior resonance control but poor vibration isolation properties.

**Steady-State Time Domain Response**  Now we show the time domain signals at two selected frequencies in the resonance control region for each hydraulic mount. Figure 4.22 depicts the steady-state responses $\ddot{x}_o(t)$, $\ddot{x}_r(t)$ and $p_1(t)$ at 8 Hz for the mount with code $N$. Note that $\ddot{x}_r$ of the plotted $\ddot{x}_r$ is 1.7 m/s$^2$ and $\ddot{x}_o$ is more regular than $\ddot{x}_r$. In the plot of $p_1$, we observe the
Figure 4.21 Measured output acceleration and relative engine displacement amplitude spectra; $\ddot{x}_e = 1.7 \text{ m/s}^2$. 
decoupler dynamics stages 3 through 6 as described in Section 3.4.2. Very small pressure ripples are exhibited during the decoupled states as in the vehicle model simulation.

In particular, even though the command voltage signal is sinusoidal, $\ddot{x}_i$ is highly non-sinusoidal due to the dynamic interaction between the heavier $m_e$ and the lighter shaker table (Doebelin, 1980) in addition to the decoupler switching dynamics. The shaker table is supported by flexure elements, whose elastic stiffness is measured to be $8.3 \times 10^4$ N/m for the given $m_e$, and damping coefficient is generally of the same order of magnitude as the suspension parameter $b_s$ given in Section 2.4.3. Note that the flexure stiffness is also of the same order of magnitude as $k_s$. Therefore we might guess at a glance that the shapes of $\ddot{x}_i(t)$ and $\ddot{x}_o(t)$ look like those of $\ddot{x}_s(t)$ and $\ddot{x}_r(t)$ of Fig.4.1 respectively. However, by comparing Fig.4.22 with Fig.4.8, we observe the converse: i.e. $\ddot{x}_i$ is similar to $\ddot{x}_s$ whereas $\ddot{x}_o$ looks like $\ddot{x}_r$. Note that the excitation signal $\ddot{x}_i(t)$ is recorded at the shaker table (analogous to a light-weight $m_s$) in this vibration testing, while the excitation in the vehicle model is given through $F(t)$ on $m_e$.

Figure 4.23 shows the response at 13 Hz for the same mount. The right-hand column depicts a detailed view of each signal. By comparing with Fig.4.22, we observe that the peak value of $\ddot{x}_o$ is increased and $\ddot{x}_i$ becomes more irregular. In addition, the pressure ripples during the decoupled state are more evident.

Figure 4.24 shows the various signals at 7 and 9 Hz for the mount with code A. We observe that $x_i$ and $x_o$ are nearly sinusoidal even though, especially, $\ddot{x}_i$ is quite irregular. In consequence, $-x$ is also almost sinusoidal at 7 Hz. Note that the shape of $\ddot{x}_o$ looks rather different from that
Figure 4.22  Measured steady-state time histories for hydraulic mount (code N, $\Delta_d=0.7$ mm) at 8 Hz; $\ddot{x}_i=1.7$ m/s$^2$. 
Figure 4.23  Measured steady-state time histories for hydraulic mount (code N, $\Delta_d = 0.7$ mm) at 13 Hz; $\ddot{x}_i = 1.7$ m/s$^2$. 
Figure 4.24  Measured steady-state time histories for hydraulic mount (code A, $\Delta_d = 0.7$ mm); $\ddot{x}_i = 1.7$ m/s$^2$. 
for code $N$ which is depicted in Figs.4.22 and 4.23. In particular, the noticeable ripples in $\dot{x}_o$ may be related with the undamped decoupler dynamics.

Figure 4.25 shows the response at 8 and 13 Hz for the mount with code $Q$. We observe that $\dot{x}_s$ and $\dot{x}_i$ look similar to those for code $N$. This makes sense because the geometric properties of code $Q$ are closer to code $N$ rather than code $A$. In particular at 13 Hz, acceleration ripples are exhibited in the decoupled state and $x_i$ appears rather triangular.

**Starting Transient Response** We now present the starting transient signals $p_1(t)$ and $\ddot{x}_i(t)$ in the vibration isolation region for the mount with code $N$. The main purpose is to qualitatively compare the measured signals with the vehicle model simulation results, and to identify the unmodeled dynamics. In this harmonic testing, the starting transients are captured with an armed trigger mode in the spectrum analyzer; see Appendix D for details. Figure 4.26 shows the captured signals at 16 Hz; Figs.4.26(c) and (d) depict the detailed views for the initial transients in Figs.4.26(a) and (b) respectively. We note that the response $p_1$ in Fig.4.26(a) looks like that in Fig.4.10. Initially the hydraulic mount experiences the coupled state, and thereafter the decoupler appears entirely open till 0.2 s. However, the experiment exhibits two major differences from the simulation. First, beyond 0.35 s, $p_1$ resumes to increase since the decoupler is closed for some duration. This may be the reason that the measured $\ddot{x}_o$ is relatively higher than the predicted $\ddot{x}_i$ in the vibration isolation region as described in Fig.4.21(a). Second, the pressure ripples which occur during the initial decoupled-state are more noticeable in the experiment. These discrepancies
Figure 4.25  Measured steady-state time histories for hydraulic mount (code $Q, \Delta_d = 0.7$ mm); $\ddot{x}_i = 1.7$ m/s$^2$. 
Figure 4.26 Measured starting transients for hydraulic mount (code \( N, \Delta_d = 0.7 \text{ mm} \)) at 16 Hz; \( \ddot{X}_i = 1.7 \text{ m/s}^2 \).
may result from fluid inertial interactions between the chambers and the decoupler. In addition, the effect of the decoupler rubber was not considered at all in the theory. The irregular peaks in $\ddot{x}(t)$ indicate the decoupler switching dynamics.

Figure 4.27 shows the starting transients at 15.25 Hz. We observe that the peak pressure decreases with time, up to 0.25 s. This characteristic was also exhibited in Fig.4.10, and it results from the fluid damping dynamics of the inertia track. However the peak pressure does not decrease anymore beyond 0.25 s. In particular Fig.4.27(c) clearly depicts that, during the decoupled state, the amplitude of the pressure ripple is attenuated by itself. This response characteristic was also observed by our theory as illustrated in Fig.3.17(b).

4.4.3 Impulse (Shock) Excitation
Recall that the ideal impulse excitation is not achievable in real-world systems. Generally, in carrying out some experiment on dynamic systems, the approximate impulsive input denotes the pulse excitation with very short duration (Doebelin, 1980). Furthermore, the shape of pulse is of no consequence in achieving the approximate impulse excitation as long as the pulse is much faster than the response speed of the tested system, say about 10 times.

Now, in this vibration testing, the shock or impulse response is obtained by applying $\ddot{x}$ of a fast-enough pulse and measuring $\ddot{x}_o$. The pulse $\ddot{x}$ may be produced by feeding a step voltage command signal to the power amplifier. In practice, since the signal generator employed in this experiment can not yield the genuine step voltage signal, its menu are set to
Figure 4.27  Measured starting transients for hydraulic mount (code N, $\Delta_d$=0.7 mm) at 15.25 Hz; $\ddot{x}_i$=1.7 m/s².
the 20 V peak-to-peak square waveform with 0.1 Hz. Note that the 0.1 Hz square wave is just the step signal in the period up to 5.0 s. Therefore, provided the impulse response of the engine-mounting system settles down in less than 5.0 s, we may practically be generating the step command signal.

**Shock Response** Figure 4.28(a) shows the recorded $\ddot{x}_i(t)$. We observe that the main pulse takes place within 2.0 ms. Figures 4.28(b)-(d) compare the rubber and various hydraulic mounts with regard to the shock response $\ddot{x}_e(t)$. Note that, as in harmonic testing, the shape of $\ddot{x}_e$ looks like $\ddot{x}_i$ depicted in Fig.4.12(b) of the vehicle model simulation. Although the hydraulic mount produces a higher first peak, it quickly damps down the subsequent responses. The effect of the different inertia track code on the shock response is not significant. In particular, even for the rubber mount, the motion of $m_r$ settles down more quickly than in the simulation. This is because the vertical motion of $m_e$ is restricted by the guide rod anyhow; it is impossible to perfectly balance the iron weights with manual work. We may now verify the rationale for the impulse response in what follows. First, the duration of $\ddot{x}_i$ is much shorter than enough with reference to the response speed of $\ddot{x}_e$. Second, the response settles down within 1.0 s.

Figure 4.29(a) depicts the relative engine displacement measured by the LVDT. In fact, it is not $x(t)$ but $-x(t)$ as denoted in the plot; recall the definition of $x(t)$ given in Eq.(4.15). This is because the spectrum analyzer was triggered at the rising edge of the square wave command signal. We observe that the unfiltered signal is quite noisy as mentioned in Section 4.4.1. Figures 4.29(b)-(d) compares the rubber and various hydraulic
Figure 4.28  Measured engine mass acceleration for shock excitation.
mounts with regard to \(-x(t)\). The shape of \(x(t)\) looks similar to that in Fig.4.12(c) of the simulation, and the hydraulic mount quickly stabilizes the engine-mounting system. The effect of the different inertia track codes on \(x(t)\) is not significant.

4.5 Discussion

By analyzing the vehicle model and carrying out vibration testing on the engine-mounting system, the resonance control, vibration isolation and shock absorption capabilities of the passive hydraulic mount were examined. In addition, two performance limitations of the regular mount were identified. First, in the resonance control region, the stiffening phenomenon is observed as in the quasilinear analysis presented in Chapter II. Beyond the engine resonant frequency, this phenomenon yields a higher sprung mass acceleration amplitude than the low-damping rubber mount. Second, in the vibration isolation region even below 20 Hz, the dynamic interaction between the fluid masses inside the decoupler control volume and upper chamber is observed in the starting transient signals measured from the vibration testing. This dynamic interaction makes the transmissibility of the engine-mounting system slightly higher than the prediction.

It was assumed in the vehicle model simulation that the engine force is harmonic and its amplitude is uniform over 3-20 Hz. The actual engine force acting on the engine mount may be measured at each engine speed by installing a force transducer at the mount location in actual engine-block testing. With the measured engine force, the vehicle model simulation may
Figure 4.29 Measured relative engine displacement for shock excitation.
provide more realistic evaluation of the passive hydraulic mount performance.

The most important outcome from the vehicle model simulation is that we have clarified the performance differences between the regular and inertia track hydraulic mounts by employing the mount nonlinear mathematical model. As far as the resonance control and shock absorption properties are concerned, the inertia track mount is superior to the regular mount. However, the vibration isolation properties of the inertia track mount is inferior to those of the regular mount or low-damping rubber mount. These observations form the basis of the adaptive hydraulic mount system as presented in the following chapter.
CHAPTER V

A NEW ADAPTIVE HYDRAULIC MOUNT SYSTEM
EMPLOYING THE ENGINE INTAKE-MANIFOLD VACUUM

5.1 Introduction

In the preceding chapters, we identified the low-frequency performance limitations and high-frequency functional problems of the regular passive hydraulic mount. The inherent nature of the regular passive mount is that the engine resonance generally arising in the vicinity of 10 Hz should actually be initiated for the mount to dissipate the engine-mounting vibration energy. In other words, fluid damping related with the inertia track flow can not be generated during the relative engine motion corresponding to the decoupler gap. On the other hand, major functional problems of the hydraulic mount are associated with the undesirable side effect of fluid inertia. As presented in Chapter II, the noise, vibration and harshness (NVH) problems of the regular mount result from the fluid resonance at higher frequencies beyond 100 Hz. Furthermore, we observed in Chapter IV that the fluid inertial effect influences the vibration isolation properties of the regular mount even below 20 Hz.

In order to resolve these shortcomings at lower frequencies and to improve the high frequency performance, a new broadband adaptive hydraulic mount system employing an on-off damping control scheme is developed in this chapter. Since the experimental vehicle setup is not
available for this study, only the functional and implementation details of the adaptive system are presented in what follows.

5.2 Description of Adaptive System

We presented in Chapters II and IV the resonance control, vibration isolation and shock absorption properties of various mount configurations. The low-damping rubber mount is best in isolating the engine disturbance force from the vehicle frame at high frequencies beyond the engine resonance. On the other hand, the inertia track mount is most appropriate in controlling the engine resonance amplitude and absorbing the shock energy. As a result, the basic idea behind this broadband adaptive system is to let the hydraulic mount function as a rubber mount for the purpose of vibration and acoustic isolations, and as an inertia track mount for the purpose of resonance control and shock absorption. Note that the high-frequency NVH problems can not arise since the adaptive mount functions as a low-damping rubber mount beyond a certain operating frequency.

This adaptive system is comprised of two modules: a mechanical actuation system and an electronic controller. The engine intake-manifold vacuum is employed as the actuation power source in implementing the on-off damping control mechanism. A discrete logic circuit or microprocessor system can be employed as the electronic control module (ECM).

5.3 Mechanical Actuation System

Figure 5.1 shows the arrangement of the adaptive system. The new hydraulic mount is equipped with rubber sheet just under the top element and does not incorporate the decoupler. The rubber sheet and the lower
rubber bellow are connected to the engine intake-manifold vacuum through two two-position three-way on-off solenoid valves which are controlled by an ECM. During the normal vehicle operation (Crouse and Anglin, 1985), the intake-manifold vacuum reaches down to 3.9-6.4 psia (17”-22” Hg vacuum). Recall that, in Fig.3.31(a), the upper chamber of the inertia track mount experiences vacuums down to 4 psia at 20 Hz where $X=1.0$ mm. However, note from Fig.4.5(c) that the vehicle model simulation for the inertia track mount yields $X=0.5$ mm even when the engine force takes the large amplitude ($F_a=500$ N). Therefore, the intake-manifold vacuum may be enough as the vacuum source for this adaptive system. Otherwise, a small vacuum pump can also be employed.

In reality, several vacuum actuators or motors utilizing the intake manifold vacuum are being installed in cars (Crouse and Anglin, 1985). For instance, to name a few, we can find their usage in automotive emission control systems: a thermostatically controlled air cleaner and an EGR (Exhaust-Gas Recirculation) valve. They are also employed in headlight-cover control systems, EFE (Early Fuel Evaporation) systems, a vacuum-advance mechanism on the distributor of a contact-point ignition system, etc.

The operating principles of the adaptive system are as follows. Provided a high damping property is desirable, for instance at lower frequencies, a vacuum pressure is applied to the upper rubber sheet while the lower bellow is open to the atmosphere. The fluid mass is pulled up and the upper rubber sheet is coupled to the top element. As a result, the adaptive mount basically becomes an inertia track hydraulic mount. Let us call this as the "hard" or "sport" state of the adaptive mount. On the other
hand, provided a low damping property is desirable, for instance at higher frequencies, a vacuum is applied below the lower bellow while the upper rubber sheet is open to the atmosphere. The fluid mass is pulled down and the upper rubber sheet is decoupled from the top element. As a result, the engine motion can not excite the fluid vibration, and the adaptive mount essentially becomes a low-damping rubber mount. Let us call this as the "soft" state of the adaptive mount.

![Diagram of the adaptive hydraulic mount system](image)

**Figure 5.1 Arrangement of the adaptive hydraulic mount system.**

### 5.4 Control Scheme

Let us call the solenoid valves controlling the vacuum lines connected to the upper rubber sheet and the lower bellow as valve 1 and valve 2 respectively as denoted in Fig.5.1. Electric control signals for the ECM are provided by
various on-board sensors: for instance, engine speed sensors such as tachometer generator and magnetic pickup, acceleration and brake sensors, etc.

When the control signals inform the ECM that the hard state is required, \textit{i.e.} during engine idle, start-up, transmission shifting, traveling on bumpy roads, abrupt acceleration or deceleration, braking, cornering, etc., the ECM turns off valve 2 and on valve 1. As a result, a vacuum is applied to the upper rubber sheet, and valve 2 is open to the atmosphere. Note that, by processing various sensor signals in this adaptive system, the ECM can activate the above-described procedure before the engine bounce takes place. On the other hand, when the ECM analyzes the control signals that the soft state is required, \textit{i.e.} during the drive on a smooth highway with the highest gear level, it turns off valve 1 and on valve 2. As a result, a vacuum is applied to the rubber bellow, and valve 1 is open to the atmosphere.

\textbf{5.5 Discrete-Logic Controller}

In what follows, let us take the engine speed as the sole control signal in order to show the basic features of the electronic valve-controller. The above-mentioned control scheme is implemented by using logic gates in this section. A microprocessor based controller is described in Section 5.6. Note that all of the electric circuits have been built on solderless breadboards and their functions have been completely tested.

\textbf{5.5.1 System Design}
We want to design an electronic controller with logic gates such that the adaptive system can operate in the "automatic" or "manual" mode. Let us employ two switches to select the operating mode as illustrated in Table 5.1; the switch value is 1 when it is "on", and 0 when it is "off". In the manual mode, the hard or soft state of the adaptive mount is determined on the basis of switch setting. On the other hand, in the automatic mode, the hard or soft state is determined on the basis of the operating engine speed. Therefore, let us employ the voltage comparator LM111 to compare the operating engine speed voltage signal $V_e$ with a reference voltage $V_{ref}$. The output from the LM111 is high when $V_e > V_{ref}$, and it is low when $V_e < V_{ref}$.

Table 5.1 Operating modes of discrete-logic valve controller.

<table>
<thead>
<tr>
<th>Switch 2</th>
<th>Switch 1</th>
<th>Operating mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1</td>
<td>Manual (hard)</td>
</tr>
<tr>
<td>1</td>
<td>0</td>
<td>Manual (soft)</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>Automatic</td>
</tr>
<tr>
<td>0</td>
<td>0</td>
<td>Automatic</td>
</tr>
</tbody>
</table>

Now we do the combinational logic design to implement all of the control functions described above. Figure 5.2 shows the logical truth table for the "mount state selector". The inputs to the state selector are $A$ (switch 1), $B$ (switch 2) and $C$ (LM111). The output from the state selector is $X$. Valve 2 turns on when $X$ is high (1), and it turns off when $X$ is low (0). The truth table is plotted in a three-variable Karnaugh map as illustrated in Fig.5.3.
The necessary design equation for the state selector is derived from the Karnaugh map as below.

\[ X = \overline{A}B + \overline{A}C + BC \]  \hspace{1cm} (5.1)

From the minimal sum of products expression given in Eq.(5.1), we note that the mount state selector can be implemented with three two-input AND gates and one three-input OR gate.

![Truth Table](image)

Figure 5.2 Truth table for the mount state selector.

![Karnaugh Map](image)

Figure 5.3 Karnaugh map for Fig.5.2.

5.5.2 Hardware Design
Figure 5.4 shows the circuit diagram of the ECM incorporating the mount state selector designed in the preceding section. Taking into account the harsh automotive environment, all TTL (Transistor-Transistor Logic) chips included coincide with the military application version 5400 series (-55°C through 125°C) which is equivalent to the standard 7400 series. The fixed-output voltage regulator LM7805C regulates the 12 VDC source of the automotive battery to the TTL-compatible 5 VDC supply. To avoid spurious signals from the single-pole double-throw (SPDT) toggle switch, a simple $\overline{SR}$ latch debounce circuit constructed from two cross-coupled 5400 NAND gates is incorporated for each switch.

The dc solenoid valves are directly operated by the 12 VDC battery. When $X$ is high, the inverter buffer/driver 5406 with open-collector high-voltage output turns on valve 2. On the other hand, when $X$ is low, the 5406 turns on valve 1 since the output from the inverter 5404 is now high. The coil resistance value should be equivalent to the pull-up resistor such that the current rating of the 5406 is not exceeded; its maximum sink current rating is about 30 mA (Texas Instruments, 1988). The silicon rectifier diode 1N4001 is placed in parallel with the valve coil in order to function as a reverse-biased protection diode. It can handle the "inductive kick" high transient voltage which is generated when the valve is turned off (Wakerly, 1990). Otherwise, the inductive transient may damage the 5406.

5.5.3 Controller using ADC and AC Solenoids
The preceding valve controller was able to use the analog voltage signals with the LM111. We now present a purely discrete valve controller in Fig.5.5, where the manual modes are not included. We observe in this
ECM (Electronic Control Module)

Figure 5.4 Schematic diagram of discrete-logic valve controller.
design that the function of the LM111 is carried out by the 8-bit magnitude comparator which is implemented by cascading two 4-bit magnitude comparator 5485s.

Since the 5485s compare two 8-bit binary (0 or 1) values, the reference and operating engine speed signals should be binary. The reference engine speed is set with the 8 position DIP (dual in-line package) switch and displayed on the 8 light-emitting diodes (LEDs). Note that the ON current per one LED is limited to about 10 mA by the 470 Ω resistor. With this configuration, the LED turns on and the switch value appears as 1 to the input P of the 5485 when the switch is closed. The analog signal representing the operating engine speed is transformed to the 8-bit digital value through the analog-to-digital converter ADC0804. In particular, the ADC0804 operates in the free-running mode and, after power-up, a momentary grounding of its WR(L) input is needed to guarantee the operation. The specifications for the ADC0804 are described in Section 5.6.2.

In this design, valves 1 and 2 are operated by ac solenoids unlike those in Fig.5.4. Therefore, the discrete logic circuits are interfaced to them through solid-state relays. Note that the solid-state relay can be operated directly by the totem-pole output from the TTL gate. Through the open-collector inverter, the two LEDs indicate which valve is turned on.

5.5.4 Bench Setup for Adaptive Hydraulic Mount System
A bench setup was prepared to demonstrate the adaptive hydraulic mount system. Figure 5.6 shows its overall view. The power supply and vacuum pump simulate the automotive battery and engine intake-manifold vacuum respectively. Figure 5.7 shows the electrical components board. Each
Figure 5.5  Schematic diagram of the purely discrete valve-controller.
Figure 5.6  Adaptive hydraulic mount system.

Figure 5.7  Electrical components board.
pneumatic valve is operated by the ac solenoid as illustrated in Fig.5.5, whereas the ECM built on the breadboard is identical to that given in Fig.5.4. Two potentiometers simulate the operating and reference engine speeds. Appendix A.7 lists the manufacturer’s specifications for equipment.

5.6 Microprocessor Based Controller

When a variety of control functions is to be implemented, the microprocessor is generally employed as a versatile design tool (Breeding, 1988). The controllers relying on the discrete logic chips as illustrated in Section 5.5 or programmable logic elements such as the field programmable gate array (FPGA), programmable logic array (PLA) and programmable array logic (PAL) are not as ubiquitous as microprocessor-based controllers.

5.6.1 System Description

Figure 5.8 shows the general organization of the microcontroller system for controlling the electro-pneumatic solenoid valves. The microprocessor unit (MPU) is the basis of the system. It is connected to memory components and input/output units or peripheral interface adaptors (PIA). Necessary information is exchanged between them through the system bus: the address, data and control buses. The read-only memory (ROM) stores the necessary program for the particular task. The read/write or random-access memory (RAM) temporarily stores the variables and stacks during the microcontroller system operation. Note that the outside world exchanges information with the system through two PIAs. The analog
Figure 5.8  Organization of the microcontroller system.
voltage signal representing the engine speed is transformed to the 8-bit digital value through ADC. The digitized signal is displayed on the LEDs0 through PIA0. Two switches select the operating mode of the adaptive hydraulic mount as in the discrete-logic controller described in Section 5.5.1: automatic or manual mode. On the basis of the switch setting, the MPU controls the operation of the on-off solenoid valves through the valve driver. Specific control functions are listed in Table 5.2, where the switch value is 1 if it is "on" and 0 if it is "off". In reality, similar operating modes can be found in the adaptive suspension systems currently installed in several models of automobile even though the functional details between the adaptive hydraulic mount and suspension systems are quite different. One example is the General Motors (GM) Skylark; they are called the driver-selectable "sport", "soft", and "automatic" modes. The second set of LEDs displays the engine speed signal through PIA1 by changing their illuminating speed.

<table>
<thead>
<tr>
<th>Switch 2</th>
<th>Switch 1</th>
<th>Operating mode</th>
<th>LEDs display 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1</td>
<td>manual (hard)</td>
<td>yes</td>
</tr>
<tr>
<td>1</td>
<td>0</td>
<td>manual (soft)</td>
<td>yes</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>automatic</td>
<td>yes</td>
</tr>
<tr>
<td>0</td>
<td>0</td>
<td>automatic</td>
<td>no</td>
</tr>
</tbody>
</table>

Before we discuss the hardware design, we need to specify how memory space is allocated. Figure 5.9 shows the memory allocation for this
microprocessor system. Note from this figure that the RAM is located in memory space from 2000H to FFFFH and the ROM's addresses run from E000H to FFFFH. As we shall see shortly, the amount of ROM and RAM space actually needed is very small. Therefore, the use of 8K ROM and 8K RAM is, in effect, overkill for this application.

![Memory Map for Valve Controller](image)

Figure 5.9 Memory map for the valve controller.

### 5.6.2 Hardware Design

Figure 5.10 shows the Motorola 6809 based microcontroller system built on a breadboard, and its circuit diagram is depicted in Fig.5.11. This circuit was built and tested by employing the HP64000 in-circuit emulation (ICE) microprocessor development system, which is located in DL505 of the Department of Electrical Engineering at The Ohio State University. Each component is described in detail in what follows.
Microprocessor-based valve controller built on breadboard.
Figure 5.11  Schematic diagram of microcontroller system.
This high density NMOS (n-channel Metal-Oxide Semiconductor) microprocessor is the basic processing unit of the system. As hardware features, the microprocessor bus consists of 8-bit data bus, 16-bit address bus and control lines. The 8 tri-state data lines are "floating" in the high impedance state if the processor is not using these lines. The 16 tri-state address lines enable the MPU to directly access 64K bytes of memory. As architectural features, the MPU has two 16-bit index registers, two 16-bit indexable stack pointers, two 8-bit accumulators that can be concatenated to form one 16-bit accumulator, and a direct page register that allows direct addressing throughout memory. As software features, the MPU has 10 addressing modes and 16-bit arithmetic is possible.

Even though it is not depicted in Fig.5.11, the microprocessor bus is buffered from the rest of the system to enhance the drive capability and protect the MPU. The address and control buses are buffered by using the line driver 74LS244 TTL chips, whereas the data bus is buffered by employing the bidirectional transceiver 74LS245 chips. In particular, pay attention to the way in which the reset circuit is connected to the MPU and the rest of the system (Breeding, 1988). All the unused pins are negated through pull-up resistors. The system clock is 0.895 MHz for our design since the crystal frequency is 3.579 MHz; the system clock is 1/4 of the crystal frequency. Manufacturer: Motorola Semiconductor Products, Inc.

This NMOS PIA is the parallel interface device which provides the universal means of interfacing peripheral equipment to the M6800 family of microprocessors. In general, peripheral device interfaces perform two functions: latching data and control. The 6821 is capable of interfacing the
MPU to peripherals through two 8-bit bidirectional peripheral data buses and four control lines. The functional configuration of the PIA is programmed by the MPU during system initialization. Each of the peripheral data lines can be programmed to act as an input or output, and each of the four control/interrupt lines can be programmed for one of several control modes. This allows a high degree of flexibility in the overall operation of the interface. Manufacturer: Motorola Semiconductor Products, Inc.

2764    The NMC27C64-150 is a high-speed 64K UV (ultra-violet) erasable and electrically reprogrammable CMOS (Complementary Metal-Oxide Semiconductor) EPROM (Erasable Programmable Read-only Memory). Its address access time ($t_{ACC}$) is 150 ns, and programming voltage ($V_{PGM}$) is 12.5 V nominally. Because the 2764 is completely static memory, no clock or timing strobe are required. The output D<7:0> are the contents of the cell whose address appears on the address lines A<12:0> only when both the chip select input CS(L) and the output enable input OE(L) are asserted low. The only two considerations in interfacing the EPROM to the MPU are to satisfy $t_{ACC}$ and to avoid data bus contention. Note, in this interface design shown in Fig.5.10, that CS is asserted prior to OE. In this case, we can think of CS as a kind of "level" signal used to enable the chip and OE as the "pulse" or timing signal to indicate when the chip is to output data to the data bus. Manufacturer: National Semiconductor Corporation.

6264    The HM6264LP is a 8Kx8-bit static CMOS RAM. The 6264 is pinout compatible with the 2764. As in the case with the 2764, addressed data
appears at the 6264's data pins when both CS(L) and OE(L) are asserted. Similarly, data placed on the 6264's data pins will be written into the RAM when CS(L) and WE(L) are asserted. Note the striking similarity between the 6264 and the 2764 in their interface designs. Manufacturer: Hitachi Limited.

ADC0804  This is the CMOS 8-bit successive approximation A/D converter that uses a differential potentiometric ladder similar to the 256R products. Differential analog voltage inputs allow increasing the common-mode rejection and offsetting the analog zero input voltage value. Its conversion time is 100 μs. Figure 5.11 depicts the interface of the free-running 0804 to the 6809 through the 6821. The end of conversion signal INTR(L) and the system reset signal RESET(L) are ANDed together into the start of conversion pin WR(L). As a result, the external WR(L) pulse is automatically generated during the first power-up cycle and thereafter the 0804 runs by itself. Manufacturer: National Semiconductor Corporation.

L293  This is a quadruple high-current half-H driver designed to provide bidirectional drive currents of up to 1 A at voltages from 4.5 to 36 V. All inputs are TTL-compatible. Each output is a complete totem-pole drive circuit with a Darlington transistor sink and a pseudo-Darlington source. Manufacturer: Texas Instruments, Inc.

In what follows, we describe the switch and valve interfacing circuits in detail. Figure 5.12 shows the switch interfacing hardware. The debounced signals from the SPDT switches 1 and 2 are connected to CA1 through the
Figure 5.12  Implementation of switch interface.
exclusive OR gate 7486. The control register A (CRA) of the PIA1 is programmed to generate an interrupt request on a low to high transition of CA1. In this way, the MPU can provide the manual mode in a minimal delay. When the MPU acknowledges the interrupt request IRQ(L), it reads the switch values through PA(0) and PA(1). The MPU determines from the switch values whether the manual mode requests the soft or hard mount states.

Figure 5.13 shows the schematic details of the valve driver circuit using the L293. In the hard state, the MPU makes PA(2) high and PA(3) low. As a result, valve 1 is turned on and valve 2 is off. On the other hand, in the soft state, the MPU makes PA(2) low and PA(3) high. Consequently, valve 1 is turned off and valve 2 is on. The high-speed output clamp diodes 1N4935 are used to suppress the inductive transient. As in another valve driver circuit given in Fig.5.4, the L293 directly employs the 12 VDC battery power. However, note that the L293 can provide much more drive currents than the 5406.

5.6.3 Software Design

With this hardware, we are now in a position to write the software necessary to carry out the control function illustrated in Table 5.2 on the basis of the memory allocation shown in Fig.5.9. Figure 5.14 gives the Cross-16 assembler output listing of the program obtained in DL417 of the Department of Electrical Engineering at The Ohio State University.

This program begins by, first, defining the stack pointers, then defining the addresses associated with the variables used in the program, and finally, defining the PIA registers. Once the address space has been set
up, the actual program, starting at location MAIN, begins by initializing the stack pointers and the PIAs. The remaining program components are generally self explanatory. In particular, the table look-up procedures are used in determining whether the operating engine speed is faster than the threshold speed and in changing the illuminating period of the LEDs display.

![Diagram](image)

Figure 5.13 Interface design for the valve driver.

The assembled program is down-loaded from the 386SX-PC to the Omnitech 6589 ROM emulator through the standard RS-232 link. The communications package used in this process is PROCOMM. The ROM emulator then programs the EPROM. In addition, the frequent task of erasing and reprogramming the EPROM was facilitated by the EPROM eraser located in CL235.
CPU "6809, TBL"
HOF "M0T8"

;**********************************************************
;** INTR **
;** SOLENOID VALUE CONTROL **
;** DATA ACQUISITION THROUGH ADC 0004 AND LED DISPLAYS **
;** INTERRUPT IRQ ENABLED **
;** PORT A: INPUT, PORT B: OUTPUT **********
;**********************************************************

3FF0 =
  SSTACK: EQU 3FF0H ;SET UP STACK POINTER
3EFF =
  USTACK: EQU 3EFFH ; AND USERS STACK PTR.
3FFE =
  PVALUE: EQU 3FFEH ;STORAGE ADDR. FOR POT VALUE
3FFC =
  PADDR: EQU 3FFCH ;STORAGE ADDR. FOR POT VALUE ADDR.
3FFA =
  DADDR: EQU 3FFAH ;STORAGE ADDR. FOR DELAY PERIOD ADD.
3FF9 =
  SWITCH: EQU 3FF9H ;STORAGE ADDR. FOR SWITCH VALUE

0000 =
  ORAO: EQU 0000H
0001 =
  CRAO: EQU 0001H
0002 =
  ORBO: EQU 0002H
0003 =
  CRBO: EQU 0003H
0004 =
  ORAL: EQU 0004H
0005 =
  CRA1: EQU 0005H
0006 =
  ORB1: EQU 0006H
0007 =
  CRB1: EQU 0007H

E000 10CE3FF0 
E004 CE3EFF

E007 7F0001
E00A 7F0003
E00D 7F0005
E010 7F0007
E013 7F0000
E016 B6FC
E01B B70004
E01B BFEE
E01D B70002
E020 B70006
E023 8604
E025 B70001
E028 B70003
E02B B70007
E02E 8607
E030 B70005
E033 1CEF

c
E035 F60000
E038 F70002
E03B 4F
E03C FD3FFE
E03F F60004
E042 C403
E044 F73FF9

;**********************************************************
;** PIA INITIALIZATION **
;**********************************************************

Figure 5.14 Valve controller program.
Figure 5.14  (continued)

```
;******************************************************************************************
;***  VALVE CONTROL
;***  IF M(SWITCH)=01H, HARD MODE
;***  IF M(SWITCH)=02H, SOFT MODE
;***  IF M(SWITCH)=03H, AUTO MODE
;***  IF M(SWITCH)=00H, AUTO MODE & NO LIGHT DISPLAY

E047 F63FF9  LDB  SWITCH
E04A C101  CMPB #01H
E04C 2715  BEQ  HARD
E04E C102  CMPB #02H
E050 2709  BEQ  SOFT

E052 FC3FFE  AUTO:  LDD  PVALUE
E055 1083007D  CMPD #007DH  ;COMPARE POT WITH 2.5 VOLTS
E059 2F08  BLE  HARD
E05B 86FB  SOFT:  LDA  #0FBH  ;SOFT MODE
E05D B70004  STA  ORA1
E060 7EE068  JMP  DNUM
E063 B6F7  HARD:  LDA  #0F7H  ;HARD MODE
E065 B70004  STA  ORA1

E068 F63FF9  DNUM:  LDB  SWITCH
E06B C100  CMPB #00H
E06D 27C6  BEQ  SCAN

;******************************************************************************************
;****  LIGHT DISPLAY ROUTINE
;****  EXTENDED INDIRECT ADDRESSING MODES ARE USED
;******************************************************************************************

E06F BEE0E8  LDX  #PTBL
E072 BF3FFC  STX  PADDR
E075 8EE0D4  LDX  #DTRB
E078 BF3F0A  STX  DADDR
E07B FC3FFE  LOOP:  LDD  PVALUE
E07E 10A39F3FFC  CMPD [PADDR]
E083 2F14  BLE  DISPLAY
E085 BE3F0A  LDX  PADDR
E088 10BE3FFC  LDY  DADDR
E08C 3002  LEAX  2,X
E08E 3122  LEAY  2,Y
E090 BF3F0A  STX  PADDR
E093 10BF3F0A  STY  DADDR
E097 20E2  BRA  LOOP

;******************************************************************************************
;****  DISPLAY FLICKERING LEDs
;******************************************************************************************

E099 C6FF  DISPLAY:  LDB  #OFFH
E09B F70006  STB  ORB1
E09E 8009  BSR  DELAY
E0A0 5F  CLRBB
E0A1 F70006  STB  ORB1
E0A4 8003  BSR  DELAY
E0A6 7EE035  JMP  SCAN

;******************************************************************************************
;****  SUBROUTINE FOR DELAY
;******************************************************************************************

E0A9 C60A  DELAY:  LDE  #10
E0AB AE9FF3FFC  DELY1:  LDX  [DADDR]
E0AF 301F  DELY2:  LEAX  -1,X
E0B1 26FC  BNE  DELY2
E0B3 5A  DECB
E0B4 26F5  BNE  DELY1
E0B6 39  RTS
```
Figure 5.14  (continued)

;********************************************
;**** INTERRUPT HANDLER
;********************************************
E0B7 F60004 HANDLER:  LDB  ORA1 ;CLEAR INTERRUPT
E0BA C403  ANDB  #03H
E0BC F73FF9  STB  SWITCH
E0BF C101  CMPB  #01H
E0C1 270B  BEQ  HARDI
E0C3 C102  CMPB  #02H
E0C5 2701  BEQ  SOFTI
E0C7 3B  RTI
E0CB 86FB  SOFTI:  LDA  #0FBH
E0C8 B70004  STA  ORA1
E0CD 3B  RTI
E0CE 86F7  HARDI:  LDA  #0FH
E0D0 B7004  STA  ORA1
E0D3 3F  RTI
;********************************************
;**** TABLE FOR DELAY PERIOD
;********************************************
E0D4 2BB275322D1BL:  DBH 2BH,082H,27H,53H,22H,0F5H,1EH,96H,1AH,37H
E0DE 15D8117A0D  DBH 1SH,0DBH,11H,7AH,0DH,1BH,08H,0CH,04H,5DH
;********************************************
;**** TABLE FOR ADC CONVERTED F01 VALUES
;********************************************
E0EB 0019003200P'BL:  DBH 00H,19H,COH,32H,00H,4BH,00H,64H,00H,7DH
E0F2 009600AF00  DBH 00H,96H,00H,0AFH,00H,0CBH,00H,0E1H,00H,OFFH
;**********
; INTERRUPT AND RESTART VECTORS
;**********
FFFF ORG  OFFFF8H
FFFF E0B7 IRQ:  DWM  HANDLER
FFFF ORG  OFFFF9H
FFFFE E000 RESET:  DWM  MAIN
0000 END
5.7 Discussion

In this chapter, only the general organization and hardware implementation were presented for the broadband adaptive hydraulic mount system which employs the engine intake-manifold vacuum. The actual performance of this adaptive system should be examined by carrying out actual vehicle testing in future study. The operational feasibility of the mechanical actuation system was partially proved through the component testing. We used a very thin rubber sheet below the top element and it punctured at 16 Hz during the component testing, where X=1.0 mm and a vacuum of 2.4 psia (25" Hg vacuum) was applied to the rubber sheet by the vacuum pump. Therefore, in making the practical adaptive system, a special manufacturing technique is required to implement the mechanical actuation scheme. This may not be any difficult task in industry.

As we illustrated, the microprocessor based valve controller is most desirable as the electronic control module. Note that the vast majority of the microprocessors manufactured in the world today go into automobiles as a command control computer. The threshold engine speed at which the switching between the hard and soft states takes place should be determined from actual vehicle testing. Needless to say, when various control signals are employed in addition to the engine speed, the hardware and software designs of the microcontroller system should be modified appropriately. In particular, note that the function of the microcontroller as enclosed in the dotted lines in Fig.5.8 can be executed by a single chip such as the Intel 8098/8398 8-bit microcontroller (Intel, 1991a). For instance, the resources of the Intel 8398 are 232 bytes of internal RAM, 8 K bytes of
internal ROM, two 8-bit and two 4-bit I/O ports, 10-bit ADC with sample and hold, pulse-width modulated output, etc. Therefore, in actual applications, only two chips are needed to organize the valve controller: the microcontroller and valve driver. A specific development system is required to program the microcontroller.

5.7.1 Comparison with Prior Designs

The adaptive hydraulic mount system reported by Graf and Shoureshi (1988) implements the continuous damping control through the electronic orifice valves and servo-hydraulic injection system. Therefore, the initial hardware setup cost is quite high and the viscosity of the hydraulic fluid may be varying under the hot engine compartment condition. The semi-active mount reported by Duclos (1987) and Ushijima (1988) utilizes the electro-viscous properties of the electro-rheological (ER) fluid which is rather expensive and requires a high voltage power supply. In addition, it should be checked whether the ER fluid can maintain the initial electroviscous properties under the severe engine compartment conditions for long-term usage. The active system reported by Hagino et al. (1986) employs the electromagnetic system and induces a high hardware setup cost.

On the other hand, the adaptive system proposed in this study employs the vacuum which already exists in the engine intake-manifold as the power source of the mechanical actuation system, and implements the on-off damping control through two solenoid valves which are much cheaper than the prior designs. Therefore, the commercial prospects of our adaptive system appear promising. In particular, all of the above-
mentioned prior systems are effective only in reducing the low-frequency vibration and noise during engine idle. However, our system is also effective at higher frequencies; it does not induce the high-frequency NVH problems.

Two other adaptive hydraulic mounts employ the vacuum as the mechanical actuation power source like our mount. The semi-active mount designed by Avon (1987) utilizes the vacuum for continuous damping control. It was shown that the dynamic properties of the adaptive mount at both lower and higher frequencies can be controlled at a minimal power consumption by adjusting the fixed-decoupler compliance with the vacuum. However, the high-frequency NVH problems can not be avoided basically because the top element excites the fluid vibration. The semi-active mount currently installed on Honda Accords implements on-off damping control by using the vacuum-operated rotary valve (Honda, 1990). It was proven that the microprocessor-controlled engine mount is quite effective in reducing the low-frequency engine-mounting noise and vibration. However, this mount was not designed for influencing the high-frequency vibration and acoustic behavior. With regard to the mechanical actuator, the rotary valve mechanism of this mount is more expensive than the rubber diaphragm of our mount or Avon's mount.

Metzeler (1991) has developed an adaptive hydraulic mount equipped with an electro-dynamic decoupling system. An electromagnetic system consisting of a permanent magnet and moving coil is attached to the decoupling diaphragm under the top element to improve the high frequency acoustic behavior. However, the fluid vibration may still be excited at higher
frequencies because the adaptive mount does not function as a purely low-damping rubber mount.

5.7.2 Future Research Areas

Several issues should be clarified in future study. For instance, (i) where to install the adaptive mount out of three or four engine mount locations under the engine-transmission block, and (ii) the appropriateness of the engine intake-manifold vacuum as the power source of the mechanical actuation system. The essence of this adaptive system is how to fabricate the top element that incorporates the actuation scheme. One idea is to make some clearance inside the top element such that it consists of upper and lower parts. However, the two parts are not separate; they are connected to each other peripherally. This design should be feasible because the top element includes rubber material. Functional description is given as follows. When a vacuum is applied to the new top element, the inside clearance disappears. Therefore, the upper and lower parts move together in exactly the same manner as the top element of the passive mount. On the other hand, when a vacuum is applied to the bottom rubber bellow, the inside clearance let the upper and lower parts do not contact each other in any case. In this way, the adaptive mount functions as a low-damping rubber mount.
CHAPTER VI
CONCLUSION

6.1 Summary and Contributions

A low-frequency nonlinear lumped-parameter mathematical model of the hydraulic mount has been developed. By analyzing the vehicle model rigorously, by carrying out laboratory vibration testing on engine mounting systems, and by introducing the quasilinear analysis on the nonlinear vehicle dynamics, the low-frequency performance limitations and high-frequency functional problems of the passive hydraulic mount were identified. A new adaptive hydraulic mount system which employs the engine intake-manifold vacuum as the mechanical actuation power source and implements the on-off damping control scheme with a microprocessor based valve controller were designed to enhance the contemporary engine-mounting technology. This comprehensive analytical and experimental study of passive and adaptive hydraulic mounts makes a number of contributions to the state-of-the-art.

First, the quasilinear analysis of the nonlinear vehicle dynamics was introduced for the first time. The feature of this method is to incorporate the dynamic stiffness spectra measured from component testing directly into the frequency response functions of the vehicle model. Therefore, by employing the quasilinear analysis as an approximate method, we can evaluate the dynamic performance of a nonlinear device at the laboratory

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site without the time-consuming task of developing its mathematical model. The basic premise is that the mechanical impedance of the nonlinear component is negligible in comparison with those of the much heavier engine and sprung masses. This implies that the quasilinear analysis predicts a more reasonable frequency response at higher frequencies. Therefore, it is more appropriate for examining the dynamic performance of an engine mount rather than that of a suspension system, since the operating frequency range of the engine mount is higher than that of the suspension system. In particular, the functional problems of the passive mount at higher frequencies were identified through the quasilinear analysis. At present, this is the one and only analytical method to examine the high frequency performance of the passive mount since the highly turbulent fluid dynamics at higher frequencies are not amenable to mathematical modeling.

Second, the high-frequency dynamic characteristics of the passive mount were studied by carrying out the high-frequency component testing on simple-orifice hydraulic mounts. It has been shown that a number of fluid mass resonances arises beyond 100 Hz. This fluid inertial effect is the "culprit" for the high frequency problems of the passive mount which deteriorate its vibration isolation properties, and it is rather inevitable as long as the top element excites the fluid vibration.

Third, two kinds of nonlinear system parameter such as chamber compliances and the steady-state inertia track fluid resistances, and the effective viscosity of glycol fluid were measured by using the specifically-designed experimental setups. The selected glycol fluid exhibits very desirable physical properties such that the hot engine compartment
environment affects the dynamic properties of the hydraulic mount only minorly.

Fourth, a nonlinear lumped-parameter mathematical model of the hydraulic mount was developed by formulating the switching mechanism of the decoupler dynamics and incorporating the measured nonlinear system parameters. The mathematical model was implemented with ACSL codes, and the frequency response characteristics were extracted by using the Fourier filter algorithm. By comparing the simulation results with the measured responses in both the time and the frequency domains, it was verified that the mathematical model is reasonably valid up to 20 Hz. However, beyond 20 Hz, some discrepancies are noticeable between theory and experiment due to the unmodeled dynamic characteristics such as the inertial effect of the upper chamber fluid and the compliance effect arising from the gas-liquid phase transformation. In particular, the inertia track mount model performs better than the regular mount model. This is because the decoupler dynamics are in fact much more complicated than our mathematical formulation for the switching mechanism. For instance, the decoupler compliance effect was not accounted for in the model. As a whole, the fluid first-order inertia track dynamics and the decoupler switching dynamics of the mathematical model reasonably represent the excitation frequency-variant and deflection amplitude-sensitive dynamic characteristics of the hydraulic mount.

Fifth, the low-frequency dynamic performances of various configurations of the passive mount were rigorously examined by analyzing the vehicle model which incorporates the nonlinear hydraulic mount, and by carrying out laboratory vibration testing on engine mounting systems.
With regard to the regular passive mount, in addition to its resonance control, vibration isolation and shock absorption capabilities being examined, performance limitations related to the upper chamber pressure buildup were identified. The vehicle model simulation provides the crucial information for the adaptive hydraulic mount design that the inertia track mount has the best resonance control capability, whereas vibration testing confirms that the low-damping rubber mount has the best vibration isolation capability beyond the engine resonant frequency bandwidth.

Finally, a new adaptive hydraulic mount system has been developed, which employs an on-off damping control scheme and is expected to provide enhanced dynamic properties as an engine mount at both lower and higher frequencies over the passive mount. The basic idea of the adaptive mount is to function as an inertia track hydraulic mount for the purpose of resonance control and shock absorption, and as a low-damping rubber mount for vibration and acoustic isolation. Because the engine intake-manifold vacuum is employed as the power source of the mechanical actuation system and two two-position three-way solenoid valves implement the on-off damping control scheme, this adaptive system looks more practical, efficient and competitive than prior designs. With regard to the electronic control module, the specific design details of the discrete logic circuit and the MC6809 based microcontroller system were illustrated. In particular, the hard, soft and automatic operation modes based on the engine speed were implemented with the ECM.

6.2 Recommendation for Future Research

Several future research areas are described below.
The first area of future research concerns the low-frequency mathematical model. (i) By installing a strain-gage type pressure transducer (for instance, Sensotec's subminiature model F with flush diaphragm) on top of the internal subassembly, the upper chamber pressure $p_{12}$ should be measured and the gas-liquid phase transformation needs to be verified. (ii) Modeling of the upper chamber fluid inerterance and the compliance effect of gas-liquid phase transformation is inevitable for better prediction of dynamic response beyond 20 Hz. However, this kind of precise modeling may be difficult in the context of the present lumped-parameter approximation. (iii) Even though the frequency effect on the inertia track flow resistance is believed not to be significant, it needs to be verified through an experimental study. However, the oscillating turbulent flow resistance is very difficult to measure with the present instrumentation technology. (iv) In order to refine the nonlinear model of the decoupler dynamics, the decoupler compliance and mass should be accounted for. Since the decoupler is made up of rubber material, some compliance effect may exist. In addition, the decoupler density is greater than the glycol fluid density; the decoupler is submerged in glycol fluid.

The second area concerns the vehicle model simulation. The nonlinear hydraulic mount can be incorporated into bicycle and four corner vehicle models in order to evaluate more realistic performance features. The road-wheel excitation can be applied to such vehicle models as well.

The third area concerns the high-frequency dynamics of the passive hydraulic mount. For a more accurate evaluation of high-frequency problems of the passive mount installed in a vehicle, a rigorous mathematical model of the high-frequency fluid dynamics is required.
beyond the quasilinear analysis. However, a practical goal of the hydraulic mount research should be focused on the development of a new mount which can resolve such high-frequency problems.

The fourth but most important area concerns the new adaptive hydraulic mount system. In order to find out the real performance of the proposed adaptive system, actual vehicle testing is mandatory. A few suggestions are listed below. (i) It should be checked whether the engine intake-manifold vacuum level is high enough to function as the power source of the mechanical actuation system. (ii) The location where to install the adaptive mount under the engine-transmission block should be determined. (iii) The top element should be fabricated so as to incorporate the mechanical actuation scheme. (iv) Other adaptive or active vibration controls need to be pursued as well.
BIBLIOGRAPHY


Breeding, K. J., 1989, Microprocessor Design Fundamentals, Department of Electrical Engineering, The Ohio State University.


National Semiconductor Corporation, 1989, General Purpose Linear Devices Databook, Santa Clara, California.


APPENDIX A

MANUFACTURER'S SPECIFICATIONS FOR
EQUIPMENT AND INSTRUMENTATION

A.1 Measurement of Chamber Compliance
A.2 Measurement of Effective Fluid Viscosity
A.3 Measurement of Fluid Flow Resistance
A.4 Measurement of Engine Mount Dynamic Properties (Site 1)
A.5 Measurement of Engine Mount Dynamic Properties (Site 2)
A.6 Vibration Testing of Engine-Mounting System
A.7 Bench Setup for Adaptive Hydraulic Mount System
A.8 Data Acquisition and Signal Analysis Equipment
A.1 Measurement of Chamber Compliance

A.1.1 Pressure Transducer

Manufacturer: Sensotec, Inc.
Model: A-10, S/N 73644
Excitation (calibration): 10VDC
Pressure range: 0-50 psig (0-345 KPa)
Strain-gage type: Bonded foil
Bridge resistance: 350Ω

Wiring code:

```
Red
+ Excitation
- Excitation
Black
```

White
+ Output
Green
- Output

Setup of strain-gage conditioner (Daytronic)

Span setting: 5V=30 psig
Output: ±5V, DC to 2 Hz

A.1.2 Strain-Gage Conditioner

Manufacturer: Daytronic Corporation
Model: 3170, S/N 639
Bridge excitation: (i) Selected with I/O connector wiring; (ii) Select regulated 5 VDC for 120 Ω bridges or transducer sensitivity > 4 mV/V; (iii) Select regulated 10 VDC for 350 Ω bridges.

Balance adjustment: will balance 1.5 mV/V initial unbalance
Span adjustment: 1 to 8mV/V, full scale
Analog output: 0 to ±5 VDC, 50% overrange, 5 mA maximum
Accuracy: 0.05% of full scale

A.1.3 Pneumatic Valves and Fittings (Clippard Minimatic)
Manufacturer: Clippard Instrument Laboratory, Inc.
Swivel hose barb: Model 15055
Quick connect assembly: Model MQC-2
Air regulator: Model MAR-1P (0-100 psig)
Reusable hose clamp: Model 5000-2
Gasket (fiber): Model 11761-3
Pipe to female adaptor: Model 15006-3

A.1.4 Pressure Gage
Manufacturer: Heise Bourdon tube, Inc.
Model: 43527
Pressure range: 0-100 psig (0-690 KPa)

A.2 Measurement of Effective Fluid Viscosity
A.2.1 Pressure Transducer
Manufacturer: Staham
Range: 0-2 psig
Excitation: 5 VDC (10 VDC max.)

A.2.2 X-Y Recorder

Manufacturer: Hewlett-Packard Company

Model: 7044B

A.3 Measurement of Fluid Resistance

A.3.1 Differential Pressure Transducer

Manufacturer: Edcliff Instruments

Model: 847342-2, S/N 4058

Type: DC to DC variable reluctance

Excitation: Internal voltage regulator regulates 24-32 VDC to 28 VDC

Pressure range: 0-75 psid (0-517 KPa)

Output: 0-5 VDC

Response time: less than 1 ms

Wiring and connector

D= +Excitation (Red); A= -Excitation (Black)

B= +Output (White); C= -Output (Green)

A.3.2 Turbine Flowmeters

Manufacturer: Flow Technology

a. Model: Omniflo FTM-5-LJ, S/N 850883

Normal flow range: 0.05-0.5 gpm (0.19-1.9 lpm)

Steady-state calibration equations with antifreeze mixture:

\[ q \text{ (gpm)} = 0.00115 f + 0.01 \quad \text{for } f < 114 \text{ cps} \]

\[ q \text{ (gpm)} = 0.00115 f + 0.00239 \quad \text{for } f \geq 114 \text{ cps} \]
where \( f \) = pulses/sec.

b. Model: FT6-8A5-LB, S/N 8602287

Normal flow range: 0.5-5.0 gpm (1.9-19.0 lpm)

Steady-state calibration equation with antifreeze mixture:

\[
q \text{ (gpm)} = 0.002403 f - 0.003119
\]

**A.3.3 Rotameter (Variable Area Flowmeters)**

Manufacturer: Omega Engineering, Inc.

Model: FL-75B

Flow range: 0.6-10.4 gpm (2.3-39.3 lpm)

Accuracy: \( \pm 2.5 \% \) full scale

Repeatability: \( \pm 0.5 \% \) full scale

Float: sharp-edge type

Pressure drop: 0.43 psi max.

**A.3.4 Frequency Counter**

Manufacturer: Beckman Instruments, Inc.

Model: EPUT 6010A, S/N 126

**A.3.5 Water Pump**

Manufacturer: Teel

Model: 1P854

Type: self-priming centrifugal pump

Maximum head: 70 ft H\(_2\)O

Electric motor: Single phase, 1 HP
A.4 Measurement of Engine Mount Dynamic Properties (Site 1)

A.4.1 Electrohydraulic Material Testing System

Manufacturer: MTS Systems Corporation
Model: 853
Feature: Longitudinal and torsional actuation

A.4.2 Dynamic Force Transducer

Manufacturer: PCB piezotronics, Inc.
Model: 208A03, S/N 5950
Output sensitivity: 10.58 mV/lbf
Force range: ±500 lbf (±2220 N)
Useful overrange (10 Volt-compression only): 1000 lbf
Maximum compression: 5000 lbf
Maximum tension: 750 lbf
Discharge time constant: 2000 sec
Low frequency response limit (-5 %): 0.0003 Hz
Polarity: compression positive
Resonance frequency: 70 KHz
Rise time: 10 μsec
Output impedance: < 100 Ω
Battery power unit: Model 480B

A.4.3 Pressure Transducer

Manufacturer: Sensotec, Inc.
Model: L/4038-02, S/N 293129
Pressure range: 0-100 psia (0-690 KPa)
Exitation: 10 VDC
Strain-gage type: Bonded foil
Bridge resistance: input=352 Ω, output= 351Ω
Calibration factor: 2.0154 mV/V at full scale with 5 VDC exc. (cal.)
Setup of strain-gage conditioner (Daytronic)
   Span setting: 3V=30 psig
   Output: ±5V, DC to 2 KHz
Wiring code: Red= +Excitation ; Black= -Excitation
            White= +Output ; Green= -Output

A.4.4 Digital Thermistor
   Manufacturer: Taylor Scientific Instruments
   Model: 9930
   Range: 68-230°F (20-110°C)
   Accuracy: ±2°F from 95° to 167°F
   Feature: Ten seconds display update rate

A.4.5 Dynamic Pressure Transducer
   Manufacturer: PCB piezotronics, Inc.
   Model: H105A03, S/N 3113 & 3114
   Output sensitivity: 19.0 mV/psi & 17.0 mV/psi
   Range: 0-100 psi
   Linearity: < 1.0 % of full scale
   Natural frequency: 250 KHz
   Rise time: 2 μsec
   Discharge time constant: 1 sec
Low frequency response (-5%) limit: 0.5 Hz
Output impedance: < 100 Ω
Acceleration sensitivity: 0.003 psi/g
Battery power unit: Model 480B
Features: (i) Hermetically sealed model
          (ii) Measure shock waves and high frequency signals

A.5 Measurement of Engine Mount Dynamic Properties (Site 2)

A.5.1 High-Frequency Testing Machine

  Manufacturer: Carl Schenck AG
  Linear actuator
    Model: PLz 7D
    Nominal force: 7 KN
    Nominal stroke: 20 mm
  LVDT: Model PFM
  Load cell: Strain-gage type

A.5.2 Piezoelectric dynamic force transducer

  Manufacturer: Schenck Trebel Corp.
  Model: 9061
  Range: 200 KN
  Sensitivity: = -4.2 pC/N
  Rigidity: 1.4x10^{10} N/m
  Natural frequency: = 45 KHz
  Capacity: = 150 pF
  Linearity: < 1 % full scale
Hysteresis: < 0.5 %
Insulation resistance: >100 TΩ

A.5.3 Piezoelectric accelerometer
Maximum measuring range: ±5000 g
Maximum transverse acceleration: 500 g
Sensitivity: 5.0 pC/g
Maximum transverse sensitivity: 3 %
Resonant frequency (mounted): 20 KHz
Rise time: 20 µs
Capacity: =165 pF

A.5.4 Charge amplifier on Euro-Card
Output voltage: ±10 V
Output current: < ±5 mV
Output impedance: 10 Ω
Insulation resistance at input: ≈100 TΩ
Freq. range (-3 dB): ≈0-10 KHz
Charge amplifier type: 5055, 5056 or 5646

A.6 Vibration Testing of Engine-Mounting System
A.6.1 Electromagnetic Vibration Testing System
Manufacturer: Unholtz-Dickie Corp.
Model: TA240D-FD208
Power amplifier: Model TA240D, S/N 720
Shaker: Model FD208, S/N 116
Capacity: 1100 lbf·sine/725 lbf random peaks, ±2 inch stroke

A.6.2 Signal Generator

Manufacturer: Wavetek, Inc.
Model: Variable phase synthesizer M650 (2 MHz)

A.6.3 Linear Guide Bearings

Manufacturer: Thomson Industries, Inc.
Model: Super Ball Bushing bearing SPB-12
Type: 0.75 inch fixed diameter pillow block
Quantity: 2

A.6.4 Guide Rod

Manufacturer: Thomson Industries, Inc.
Model: Solid hardened-and-ground 60 case steel shaft
Nominal diameter: 0.75 inch
Length: 2 ft 8 in
Rockwell hardness: 60-65 C
Quantity: 2

A.6.5 Accelerometers

Manufacturer: Bruel & Kjaer Instruments, Inc.
Reference charge sensitivity at 50 Hz, 100 m/s²
a. Type 4393, S/Ns 1281485 & 1342468
Charge sensitivity: 0.319 & 0.315 pC/ms² (3.09 pC/g)
Maximum transverse sensitivity: 4.0 & 1.9 %
Typical undamped natural frequency: 85 KHz

Typical transverse resonance freq. (with mounting stud): 55 KHz

b. Type 4371, S/Ns 1270413 & 1298808
   Charge sensitivity: 0.991 & 0.997 pC/ms²
   Maximum transverse sensitivity (at 30 Hz, 100 m/s²): 2.6 & 3.1 %
   Typical undamped natural frequency: 48 KHz
   Typical transverse resonance freq. (with mounting stud): 15 KHz

c. Type 4381, S/Ns 984967 & 1308783
   Charge sensitivity: 10.03 & 10.04 pC/ms²
   Maximum transverse sensitivity: 2.4 & 0.6 %
   Typical undamped natural frequency: 25 KHz
   Typical transverse resonance freq. (with mounting stud): 5 KHz

A.6.6 Charge Amplifiers

Manufacturer: Bruel & Kjaer Instruments, Inc.
Model: Type 2635, S/Ns 853943 & 1262274
Features: (i) Built-in integrators for displacement and velocity
          (ii) High sensitivity up to 10 V/pC
          (iii) Switchable low and high frequency limits

A.6.7 Digital Filter

Manufacturer: Frequency Devices
Model: 9002
Features: (i) Dual channel
          (ii) Programmable functions
Factory channel setting
Ch #1: HP01 (8-pole 6-zero elliptic high pass filter)
Ch #2: LP01 (8-pole 6-zero elliptic low pass filter)

**A.6.8 Linear Variable Differential Transformer (LVDT)**

Manufacturer: Sensotec, Inc.
Model: MS2-200, S/N 986
Type: DC-DC
Stroke range: ±0.2 inches
Excitation: 10 VDC, no voltage regulator installed
Output sensitivity: 3.739 V/inch (=0.1472 V/mm) at 10 VDC excitation
Unfiltered output impedance: 100 Ω
Output load (min.): 20 KΩ
Noise (filtered output): 15 mV max peak to peak
Linearity on calibration (20°C, 22 KΩ load)

**A.6.9 Power Supply**

Manufacturer: System Research Corporation
Model: 3564
Input: 105-125 VAC
Output: 0-25 VDC, 0-200 mA

**A.7 Bench Setup for Adaptive Hydraulic Mount System**

**A.7.1 Vacuum Pump**

Manufacturer: Gast Manufacturing Corporation
Model: DOA-104-A4, S/N 0983
Max. vacuum: 640 mm (25") Hg
Electric motor: 115 VAC, 60 Hz, 4 A

A.7.2 Pneumatic Valve

Manufacturer: SMC Pneumatics Inc.
Model: NVF110-3G-M5
Type: 3 port/3 way 110 VAC solenoid operated valve, 10-32 port
Quantity: 2

A.7.3 Solid-State Relay

Manufacturer: Potter & Brumfield
Model: SSR-240D25
Input voltage: 3-32 VDC
Max. switching Rating/Output: 0.05-25 A rms
Features: Inverse parallel silicon-controlled-rectifier (SCR) output controlled by an opto-electronic coupler

A.7.4 Power Supply

Manufacturer: Fujitsu Denso Ltd.
Model: B14L-5105-0100A
Maximum output current: 7.5 A
Features: Regulated triple-output of +24 V, +5 V and -12 V

A.8 Data Acquisition and Signal Analysis Equipment

A.8.1 Dynamic signal analyzer

Manufacturer: Hewlett-Packard Company
Model: 35665A
Frequency range: 102.4 KHz for one channel mode
51.2 KHz for two channel mode

Frequency resolution: frequency span/400

Input ranges (full scale): 3.99 mV_{pk} to 31.7 V_{pk}

Sampling period: 3.815 µs to 2 s for one channel mode
7.629 µs to 4 s for two channel mode

Time record length: 1024 samples
Input impedance: 1 MΩ±10 %; ≤100 pF

A.8.2 Digital oscilloscope

Manufacturer: Hewlett-Packard Company
Model: 54501A
Repetitive bandwidth: 100 MHz
Vertical resolution (A/D): 8 bits
Sampling rate: 10 megasamples/sec
Features: (i) 4 channel input and display
(ii) 4 nonvolatile waveform memories
(iii) 2 volatile pixel memories
APPENDIX B

DATA TRANSFER FROM HP 35665A TO VAX 8500

The experimental data measured from component testing on the electrohydraulic testing system and from vibration testing on the electrodynamic shaker are transferred from the HP 35665A dynamic signal analyzer to the VAX 8500 main frame computer as illustrated in Fig.B.1. Its detailed procedures are as follows.

![Diagram](image)

Measured signals

Figure B.1 Schematic diagram of data handling.

1. Power on the HP 35665A.
2. Press [Save/Recall], [DEFAULT DISK] AND [INTERNAL DISK] subsequently so as to select the analyzer's internal disk as the default drive.
3. Insert a 3.5" floppy disk into the analyzer's internal disk drive. If the floppy disk is already formatted, go to step 5.
4. Press [Disk Utility] and [FORMAT DISK] subsequently, and format the blank disk with the DOS format.
5. Acquire all of the measured signals and store them on the formatted disk.

6. Now we are ready to transfer the acquired data to an IBM PC compatible (AT or higher) microcomputer by using the Standard Data Format (SDF) Utilities as a communication protocol. The SDF Utilities is a group of MS-DOS programs and should be installed in your directory on the PC's hard disk. Insert the floppy disk into the PC's floppy disk drive, say drive B.

7. Type the following command in your directory.

   SDFTOML  B:filename  /X

   Now, with regard to the time domain data, a full-precision binary MATLAB format file TIME.MAT is created.

8. Process TIME.MAT with an M-file in the environment of PC-MATLAB so as to create data file(s) which are to be transferred to VAX 8500. A sample M-file is given as program #7 in Appendix F.

9. Use PROCOMM as the communications package between the 386-SX PC and VAX 8500. Now the measured time domain signal is available in the VAX environment.
APPENDIX C

FREQUENCY RESPONSE OF CHARGE AMPLIFIER DISPLACEMENT MODE

The principle of the piezoelectric accelerometers is to generate the electric charge in proportion to the acceleration of the measuring spot and the sensitivity of the employed piezoelectric material as well. The electric charge is transformed to the more meaningful voltage signal through the charge amplifier.

Any acceleration signal may be integrated once or twice to yield the velocity or displacement signals, provided the integrating electronic circuit coincides with the "perfect" integrator. The "standard" or "perfect" op-amp integrating circuit (Fig.C.1) is generally used in analog computer applications, where the drift is "self-correcting" in conjunction with the "computing loop" that occurs in solving a differential equation. However it is rarely employed in measurement instrumentation applications because the drift or DC bias are no longer "self-correcting" without the "computing loop". The drift can make the measurement inaccurate or, even worse, cause the saturation of the "perfect" integrating circuit in long-term operation; note from the transfer function given in Fig.C.1 that its gain is infinite at DC. Furthermore the inherent op-amp imperfections may not be entirely eliminated even though high-quality op-amps are used.

Therefore, in measurement instrumentation applications, an approximate integrator is generally employed to resolve the drift problem.
Figure C.1  Electronic circuit for standard integrator.

Figure C.2  Electronic circuit for approximate integrator.
Figure C.2 illustrates an example of the approximate integrator. As shown in its transfer function, the basic idea behind the approximate integrator is to make its gain be zero at DC. Therefore it can eliminate the drift problem.

However, as in all engineering design, the approximate integrator is not ideal for all situations. In particular, two undesirable aspects of the approximate integrator keep it from integrating any kind of acceleration signals. First, its gain is less than that of the standard integrator near the frequency defined by

$$\omega_0 = 1/\tau$$  \hspace{1cm} (C.1)

Therefore, provided the signal has the frequency around \( \omega_0 \), its integrated response is attenuated inevitably. Second, the phase response is not uniform around \( \omega_0 \) unlike the perfect integrator. As a result, the periodic signals containing multiple frequencies around \( \omega_0 \) and particularly the transient signals should not be integrated by using the approximate integrator. Only a sinusoidal or single-frequency signal is recommended to integrate through it.

Now, we examine the frequency attenuation effect of the 1.0 Hz-displacement mode of the B&K charge amplifier whose specifications are listed in Appendix A.6.6. Note that \( \omega_0 \) (30% attenuation) of the approximate integrator for this displacement mode is slightly lower than 1.0 Hz (10% attenuation nominally). It is obvious from the preceding explanation that the integrated signal is attenuated around 1.0 Hz.

The purpose is to obtain the "reasonable" displacement signal from the measured acceleration signal by compensating the frequency attenuation effect of the approximate integrator. As presented in Section 4.4, the lowest frequency of concern is 3 Hz. The frequency response of the
1.0 Hz-displacement mode is calibrated by employing the arrangement shown in Fig.C.3. The calibration procedure is as follows:

Figure C.3 Experimental setup for charge amplifier calibration.

1. Install an accelerometer on the shaker table by using a lightweight mounting fixture without any other heavy mass.
2. Set the charge amplifier to the 1.0 Hz-displacement mode.
3. Sinusoidally excite the shaker at frequency $f$ and monitor the displacement signal $\ddot{x}$ through the digital oscilloscope.

$$\ddot{x}(t) = \ddot{x} \sin(2\pi ft)$$  \hspace{1cm} (C.2)

where $\ddot{x}$ is the displacement amplitude. Adjust the gain-setting knob of the vibration testing system such that $\ddot{x}=1.0$ mm. We use the tilde symbol ($\sim$) because this displacement signal measured through the charge amplifier is not the true but "attenuated" response.

4. Calculate the attenuated acceleration amplitude $\dddot{x}$ defined in the following

$$\dddot{x}(t) = -(2\pi f)^2 \dddot{x} \sin(2\pi ft) = -\dddot{x} \sin(2\pi ft)$$ \hspace{1cm} (C.3a)

$$\dddot{x} = (2\pi f)^2 \dddot{x}$$ \hspace{1cm} (C.3b)
5. Switch the charge amplifier setting to the 0.2 Hz-acceleration mode and measure the almost "true" acceleration $\ddot{x}$ given in the following.

$$\ddot{x}(t) = -\ddot{X} \sin(2\pi f) = -(2\pi f)^2 X \sin(2\pi f) t ; \quad \ddot{X} = (2\pi f)^2 X$$  \hspace{1cm} (C.4a,b)

where $\chi$ is the almost "true" displacement amplitude denoted with regard to the acceleration amplitude $\ddot{X}$. Note that $\omega_0$ of this acceleration mode is even lower than 0.2 Hz (10%). Therefore, the measured $\ddot{x}$ in the frequency range 3 to 20 Hz may be an almost "true" response.

In practice, we display $\ddot{x}$ on the digital oscilloscope via the digital filter as illustrated in Fig.C.3 because the signal $\ddot{x}$ is quite noisy at lower frequencies. Figure C.4 compares the filtered and original $\ddot{x}$ signals at 3, 5, 10 and 20 Hz. The cutoff frequency of the low-pass filter is set 10 times higher than the signal frequency for each individual $f$. Therefore, the gain for the frequency response of the filter itself is nearly unity at each $f$ whereas its phase lag can not be avoided as shown in Fig.C.4. Note, however, that we are not calibrating the phase response but just the amplitude response of the displacement mode.

At lower frequencies, the electric charge generated from the piezoelectric accelerometer is quite a small amount since the vibration level is limited in practice. The charge amplifier amplifies both response and stray noise signals. Therefore the noise signals are noticeable up to 10 Hz. In particular, the filtered $\ddot{x}$ signals are quite sinusoidal even at 3 Hz since the high-capacity electrodynamic shaker is excited without any significant load.

5. We may now find $X$ from Eqs.(C.3b) and (C.4b) as below.

$$X = \ddot{X} (\ddot{X}/\ddot{X}) = \ddot{X}/\ddot{X}$$  \hspace{1cm} (mm)

Recall that we are using $\ddot{X} = 1.0$ mm in this calibration.
Figure C.4  Unfiltered and filtered acceleration signals of the shaker table.
Table C.1 lists the calibration result for the charge amplifier of serial number 1262274. Its output gain is set to 1000 mV/(unit out) during the calibration up to 10 Hz. The frequency attenuation effect of the 1.0 Hz-displacement mode is compensated for by multiplying the acquired response \( \tilde{X} \) by the corresponding calibration factor at the operating frequency. In particular, the calibration factor is nominally taken to be 1.0 beyond 8 Hz in the vibration testing. The calibration factors for the charge amplifier of serial number 853943 are presumed to be identical to those listed in Table C.1. In addition, when different vibration levels are employed, i.e. \( \tilde{X} = 0.5 \) and 2.0 mm, the calibration factors remain nearly the same as it should be for the linear system response.

Table C.1 Calibration results for the 1.0-Hz displacement mode of the charge amplifier.

<table>
<thead>
<tr>
<th>( f ) (Hz)</th>
<th>( \tilde{X} ) (mm)</th>
<th>( \tilde{X} ) (V)</th>
<th>( \ddot{X} ) (V)</th>
<th>( X ) (mm)</th>
<th>Cal. factor</th>
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</table>
APPENDIX D

MEASUREMENT OF STARTING TRANSIENT RESPONSE IN VIBRATION TESTING SYSTEM

We presented the starting transient responses at 16 and 15.25 Hz in Figs.4.26 and 4.27 respectively for harmonic excitation where $\ddot{x}_r=1.7 \text{ m/s}^2$. Note that the amplifier gain of the vibration testing system is to be increased from the zero value. Therefore, in order to make $\ddot{x}_r$ be $1.7 \text{ m/s}^2$ instantly, we need a special operation technique as described below.

1. First set the signal generator to yield a sinusoidal command voltage with frequency of 16 Hz for example.
2. Adjust the amplifier gain such that $\ddot{x}_r=1.7 \text{ m/s}^2$ by monitoring it through the spectrum analyzer.
3. Change the frequency setting of the signal generator such that 0.1 Hz sinusoidal command is generated from it. Now the shaker may vibrate at 0.1 Hz, but $\ddot{x}_r<<1.7 \text{ m/s}^2$ since the amplifier gain has not been modified.
4. Change the frequency setting of the signal generator back to 16 Hz and press its EXECUTE button. Now the shaker may vibrate at 16 Hz with $\ddot{x}_r$ reaching $1.7 \text{ m/s}^2$ instantly.

The starting transient signals $\ddot{x}_r(t)$ and $p_1(t)$ are acquired by using the trigger function of HP35665A Dynamic Signal Analyzer. First, we
present the preliminary setup procedure for the analyzer to operate in its trigger function. In the following, by referring to the Operator’s Reference manual (Hewlett-Packard, 1991b), the hardkey in the front panel is denoted by [Hardkey] whereas its softkey is named by [SOFTKEY].

1. Power on the analyzer.
2. Press [Inst Mode] in the "measurement" group.
3. Select [HISTOGRAM/TIME] and [2 CHANNEL].
5. Select [UPPER/LOWER].
7. Select [UNFILTERED TIME CH 1] for the upper trace and [UNFILTERED TIME CH 2] for the lower trace by using [Active Trace] in the "display" group.

Let us continue to set the record length.

1. Press [Freq] in the "measurement" group.
2. Press [RECORD TIME].
3. Rotate the knob in the "marker" group until the record time is set to 1.0 s for example. Note that the [SAMPLE TIME] is automatically determined once the [RECORD TIME] is set. There are 1024 samples in one record time for the 2 channel operation whereas 2048 samples for the single channel operation.
Now let us do the following procedure to setup the trigger menu specifically.

1. Press [Trigger] in the "measurement" group. It provides trigger and arming choices.

2. Select [CHANNEL 1 TRIGGER] and press [TRIGGER SETUP].

3. Set the trigger menu as below.
   - [LEVEL]=50% (of the current input range)
   - [SLOPE]=NEGative
   - [CHANNEL 1 DELAY]=0 sec
   - [CHANNEL 2 DELAY]=0 sec

4. Press [RETURN].

5. Press [ARM SETUP].

6. Select [MANUAL ARM] and press [RETURN].

   Everything is now ready to acquire the starting transient signals. Connect $\ddot{x}$, and $p_1$ signals to the analyzer's input channels 1 and 2 respectively through the BNC connector. Excite the shaker at 0.1 Hz as mentioned above. Press [ARM] of the analyzer. At this moment, the analyzer may not be triggered because the level of $\ddot{x}$ is less than the preset trigger level; otherwise, the input range of channel 1 should be adjusted such that the analyzer is not triggered for the 0.1 Hz sinusoidal excitation. Now change the excitation frequency back to 16 Hz and press the signal generator's EXECUTE button. The analyzer may be triggered and the signals acquired.
The same procedure as described above can be applied to acquire the starting transient signals at other excitation frequencies.
APPENDIX E

IMPLEMENTATION OF SWITCHING MECHANISM WITH ACSL

In solving the ordinary differential equations numerically, the introduction of discontinuities into an otherwise continuous event (time or state events) always induces a trouble. The variable step algorithms reduce the numerical integration step size in the region of the discontinuity at the expense of increased processing time, and it is debatable how well the error control mechanism works in the event of a discontinuity in one of the state variables. Nature has no real discontinuities given a sufficiently fine time scale; discontinuities arise from simplifying the model and observing the system behavior in a macro time scale.

Note that, in real-world systems, the state variables can not be predicted ahead of time at the point of discontinuity because every physical system is causal or nonanticipative. However, such prediction is possible in the environment of numerical simulations. In line with this reasoning, the ACSL provides the SCHEDULE operator which functions as an event-finder and is very efficient computationally. Almost all kinds of switching mechanism can be implemented by a skillful application of the SCHEDULE operator.

Algorithm for Decoupler Switching Dynamics

The following ACSL codes are included in the programs of Appendix F so as to implement the decoupler switching mechanism which is described in Section 3.4.
SELP = RSW(PRES, P1-P2-ε, P2-P1-ε)
SELD = RSW(CPLD, SELP, ABS(QV3)-GAP)

SCHEDULE  MOUNT. XP. SELD

Governing equations for the hydraulic mount dynamics

DISCRETE MOUNT

Block for servicing each event

In these codes, RSW is the switch operator for real values, and PRES and CPLD are the logical constants defined as below.

PRES = .T.  for \( p_1 < p_2 \);  \quad \text{PRES = .F.  for } p_1 > p_2

CPLD = .T.  for the coupled state

CPLD = .F.  for the decoupled state

The variable GAP is initially \( V_{gap}/2 \), and it is switched to \( V_{gap} \) once the decoupler is closed or CPLD=.T. The variable QV3 denotes \( q_d \), and \( \varepsilon \) is a small real number employed to facilitate the digital computation. In this problem, \( \varepsilon=100 \) (Pa) works well.

The source codes can be explained in text as follows:

1. Provided the decoupler is open or CPLD=.F., it gets closed at the instant \( |q_d|=\text{GAP} \).

2. Provided CPLD=.T., and
(1) \( p_1 < p_2 \) or PRES=.T., the decoupler is open at the moment \( p_1 > p_2 + \epsilon \).

(2) \( p_1 > p_2 \) or PRES=.F., the decoupler is open at the moment \( p_2 > p_1 + \epsilon \).

For other implementation details, the self-explanatory programs and ACSL manual should be referred to.
## APPENDIX F

### COMPUTER PROGRAMS

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PROGRAM TIME DOMAIN SIMULATION OF REGULAR HYDRAULIC MOUNT

DESCRIPTION
This program simulates the time domain response of the regular hydraulic mount in the environment of component testing. Two inertia track paths and decoupler are modeled. Upward motion is taken positive.
If DGAP < 0.1 mm, the mount is taken as the inertia track mount.

LIST OF VARIABLES AND PARAMETERS
ACODEC : area code for decoupler control volume in decoupled state
ACODED : area code for decoupler control volume in coupled state
ACODE1 : reducer area code
AD : decoupler area (m²)
AI1 : cross-sectional area of inertia track path 1 (m²)
AI2 : cross-sectional area of inertia track path 1 (m²)
AI3 : cross-sectional area of decoupler control volume (m²)
AMP : amplitude of excitation displacement (mm)
BR : damping coefficient of top element (N·s/m)
DGAP : decoupler gap (mm)
DP : equivalent piston diameter of top element (m)
FT : transmitted force (N)
GAM : specific heat ratio
HZ : excitation frequency (Hz)
II1 : fluid inerterance of inertia track path 1 (Kg/m²)
II2 : fluid inerterance of inertia track path 2 (Kg/m²)
II3 : fluidic inerterance of decoupler control volume (Kg/m²)
KR : elastic stiffness of top element (N/m)
LAIR : column length of initial gas volume (m)
LCODEC : length code for decoupler control volume in decoupled state
LCODED : length code for decoupler control volume in coupled state
LCODE1 : inertia track code
L11 : length of inertia track path 1 (m)
L12 : length of inertia track path 2 (m)
L13 : length of decoupler control volume (m)
MU : absolute or dynamic viscosity of glycol fluid (Kg/m·s)
NCYCLE : number of excitation cycles
PATM : atmospheric pressure (Pa)
PDROP1 : pressure drop through inertia track path 1 (Pa)
PDROP2 : pressure drop through inertia track path 2 (Pa)
PDROP3 : pressure drop through decoupler (Pa)
PFAC : number of communication times per cycle
PRE : preload (N)
P0 : pressure at static equilibrium under a given preload (Pa)
INITIAL
"OPEN DATA LOG FILES"

CALL ASS

" LCODE1=1 FOR CHANNEL CODE A "
" LCODE1=14 FOR CHANNEL CODE N "
" LCODE1=18 FOR DUAL CHANNEL DEVICE "

LOGICAL CPLD, PRES
INTEGER ICODE1, LCoded, LCODEC
CONSTANT PHI=3.141592, RHO=1059., MU=3.17E-3
CONSTANT DP=.08, GAM=1.0, LAIR=0.0005
CONSTANT AD=.0023, DGAP=0.7
CONSTANT HZ=15.,NCYCLE=5., PFAC=50., AMP=1.0
CONSTANT PRE=1200., PATM=101300.
CONSTANT LCODE1=14, ACODE1=2.0
CONSTANT LCODED=19, ACODED=5., LCODEC=19, ACODEC=6.

TABLE KRTB1,1,22 ...
    /1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11., ...
    12., 13., 14., 15., 20., 25., 30., 35., 40., 45., 50., ...
    277.,278.,280.,281.,281.,282.,283.,283.,284.,284., ...

TABLE BRTB1,1,22 ...
    /1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11., ...
    12., 13., 14., 15., 20., 25., 30., 35., 40., 45., 50., ...
    1540.,779.,521.,396.,322.,273.,237.,211.,191.,174.,160., ...
    149.,140.,132.,125.,98.,88.5,79.9,74.,69.3,67.,64.5/

" INITIAL CONDITIONS: P1BAR(PA), V2BAR(IN**3), V1BAR(CC)"
" V1BAR IS TRANSFORMED TO IN**3 LATER"
TABLE PBAR.1,19 ... 
/200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
1900., 2000., ...
101590.7, 102086.1, 102867.7, 103917.7, 105236.7, ...
106762.8, 108437.3, 110236.0, 112113.7, 114136.2, ...
116405.6, 119129.9, 122604.5, 127144.8, 133094.8, ...
140218.9, 148879.0, 158735.6, 169738.0/

TABLE V1BAR.1,19 ...
/200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
1900., 2000., ...
.1038, .1704, .2482, .3333, .425, .5192, .6133, .7069, ...
.7986, .8922, .992, 1.1062, 1.2448, 1.4164, 1.6289, ...
1.8693, 2.1458, 2.4445, 2.7624/

TABLE V2BAR.1,19 ...
/200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
1900., 2000., ...
.304, .454, .602, .746, .89, 1.032, 1.172, 1.312, 1.45, ...
1.588, 1.724, 1.856, 1.982, 2.1, 2.21, 2.306, 2.394, ...
2.472, 2.542/

" CHANNEL CODE= A B C D E F G H I J K L M N O P Q DUAL DEC PL"
"   LCODE= 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19"
" MM UNIT"

TABLE LENGTH.2,19,8 ...
/1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11., 12., 13., ...
14., 15., 16., 17., 18., 19., ...
0., 1., 2., 2.5, 3., 4., 5., 6., ...
25.9, 36.7, 47.5, 58.2, 69.0, 79.8, 90.6, 101.4, 112.2, ...
123.0, 133.7, 144.5, 155.3, 166.1, 176.9, 187.7, 198.5, ...
166.1, 5., ...
25.5, 36.1, 46.8, 57.4, 68.0, 78.7, 89.3, 99.9, 110.5, ...
121.2, 131.8, 142.4, 153.1, 163.7, 174.3, 184.9, 195.6, ...
163.7, 5., ...
25.1, 35.6, 46.1, 56.5, 67.0, 77.5, 88.0, 98.4, 108.9, ...
119.4, 129.9, 140.3, 150.8, 161.3, 171.7, 182.2, 192.7, ...
161.3, 5., ...
24.9, 35.3, 45.7, 56.1, 66.5, 76.9, 87.3, 97.7, 108.1, ...
118.5, 128.9, 139.3, 149.7, 160.1, 170.5, 180.8, 191.2, ...
160.1, 5., ...
24.8, 35.1, 45.4, 55.7, 66.0, 76.3, 86.6, 97.0, 107.3, ...
117.6, 127.9, 138.2, 148.5, 158.8, 169.2, 179.5, 189.8, ...
158.8, 5., ...
24.3, 34.4, 44.5, 54.6, 64.7, 74.8, 84.9, 95.0, 105.1, ...
115.2, 125.3, 135.4, 145.5, 155.6, 165.7, 175.8, 185.9, ...
155.6, 5., ...
5., 5., 5., 5., 5., 5., 5., 5., 5., ...
5., 5., 5., 5., 5., 5., 5., ...
6., 6., 6., 6., 6., 6., 6., 6., 6., ...
6., 6., 6., 6., 6., 6., 7./
" AREA REDUCER = FULL 1.0 2.0 2.5 3.0 4.0 5.0 6.0 "
" ACODE = 0. 1.0 2.0 2.5 3.0 4.0 5.0 6.0 MM**2 UNIT"

TABLE DIA, 1, 0 ...
   /0., 1.0, 2.0, 2.5, 3.0, 4.0, 5.0, 6.0, ...
   7.501, 6.746, 5.891, 5.409, 4.906, 3.772, 30., 0.1 /

   AP = PHI*DP*DP/4.
   KR = KRTBL(HZ)*1.E3
   BR = BRTBL(HZ)
   XAMP = AMP*0.001
   P0 = PBAR(PRE) $ V20 = V2BAR(PRE) $ V10 = V1BAR(PRE)/2.54**3.

" GEOMETRIES FOR TWO INERTIA TRACKS"
LLCD1 = LCODE1 $ L11 = LENGTH(LLCD1, ACODE1)*1.E-3 $ L12 = 0.2115-L11
DL1 = DIA(ACODE1)*1.E-3 $ DL2 = DL1
AI1 = PHI*DL1**2./4. $ AI2 = AI1

" DECOUPLER GEOMETRY"
LLCD3 = LCODED $ L13 = LENGTH(LLCD3, ACODED)*1.E-3
DL3 = DIA(ACODED)*1.E-3
AI3 = PHI*DL3**2./4.

" VAIR0 = INITIAL GAS VOLUME; JUGGLE WITH LAIR TO CHANGE VAIR0"

VAIR0 = LAIR*AP
V1 = V10 $ V2 = V20
Q1 = 0. $ Q2 = 0. $ Q3 = 0. $ P1 = P0 $ P2 = P0
OMEGA = HZ*2.*PHI
CPLD = .FALSE. $ PRES = .FALSE.
VGAP = AD*DGAPE**0.001 $ XKEEP = 0. $ GAP = VGAP/2. $ X = 0.
PERIOD = 1./HZ $ CINT = PERIOD/PFAC
TF1 = NCYCLE*PERIOD-PERIOD*0.25
TF2 = NCYCLE*PERIOD
TF = NCYCLE*PERIOD-CINT
TS1 = TF2-PERIOD $ TS2 = TF2-PERIOD*2.

END "$ OF INITIAL"

DYNAMIC

DERIVATIVE

SELX = RSW(PRES, P1-P2-1000., P2-P1-1000.)
SELX = RSW(CPLD, SELX, ABS(QV3)-GAP)
SCHEDULE MOUNT XP.SELD

X = XAMP*SIN(OMEGA*T)
XDOT = XAMP*OMEGA*COS(OMEGA*T)

PROCEDURAL (P1, P2, P1E, P2E, P1EG, P2EG, PDROP1, PDROP2 = V1, V2, Q1, Q2, VAIR)

P2EG = 0.83420*V2**2.5-0.25117*V2**6+0.159415*V2**6.5
P2 = P2EG*6894.41+PATM
IF (P2.1.E.0.) P2 = 0.
P2E = P2/6894.41
IF (V1.L.E.0.) GO TO 100
PIEG = 76.86729*V1+180.81844*V1**2+(7/6.)*19.46422*V1**2.5
P1=PIEG*6894.41+PATM
IF(P1.LE.0.) P1=0.
GO TO 110
100.. CONTINUE
   P1=PATM*(VAIR0/VAIR)**GAM
110.. CONTINUE
   P1E=P1/6894.41

"PRESSURE DROP ASSOCIATED WITH TURBULENT FLOW THROUGH CHANNELS"

GO TO (1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18), LCODE1
GO TO 20
1.. CONTINUE
   PDROP1=0.634*1.732E12*ABS(Q1)**2.
   GO TO 20
2.. CONTINUE
   GO TO 20
3.. CONTINUE
   GO TO 20
4.. CONTINUE
   GO TO 20
5.. CONTINUE
   GO TO 20
6.. CONTINUE
   GO TO 20
7.. CONTINUE
   GO TO 20
8.. CONTINUE
   GO TO 20
9.. CONTINUE
   GO TO 20
10.. CONTINUE
    GO TO 20
11.. CONTINUE
    GO TO 20
12.. CONTINUE
    GO TO 20
13.. CONTINUE
    GO TO 20
14.. CONTINUE
    PDROP1=1.16*1.732E12*ABS(Q1)**2.
    PDROP2=1.99*1.732E12*ABS(Q2)**2.
    GO TO 20
15.. CONTINUE
    PDROP1=1.10*1.732E12*ABS(Q1)**2.
    PDROP2=1.99*1.732E12*ABS(Q2)**2.
    GO TO 20
16.. CONTINUE
    PDROP1=1.10*1.732E12*ABS(Q1)**2.
    PDROP2=1.99*1.732E12*ABS(Q2)**2.
    GO TO 20
17.. CONTINUE
    GO TO 20
18.. CONTINUE
    PDROP1=1.10*1.732E12*ABS(Q1)**2.
    PDROP2=0.634*1.732E12*ABS(Q2)**2.
CONTINUE
PDROP3=(Q3/0.6/A13)**2.*RHO/2.

END $ "OF PROCEDURAL"

" FLOW THROUGH INERTIA TRACK PATH 1"
REH1=D11/MU*SQR(2.*PDROP1*RHO)
II1=RHO*(LI1)/A11
Q1DOT=(P2-P1-PDROP1*SIGN(1.0, Q1))/II1
Q1=INTEG(Q1DOT,0.) $ Q1P=Q1*1.585E4
QV1=INTEG(Q1,0.)

" FLOW THROUGH INERTIA TRACK PATH 2"
REH2=D12/MU*SQR(2.*PDROP2*RHO)
II2=RHO*(LI2)/A12
Q2DOT=(P2-P1-PDROP2*SIGN(1.0, Q2))/II2
Q2=INTEG(Q2DOT,0.) $ Q2P=Q2*1.585E4
QV2=INTEG(Q2,0.)

" FLOW THROUGH DECOUPLER"
RE3=4.*RHO*ABS(Q3)/PHI/D13/MU
II3=RHO*(LI3)/A13
Q3DOT=(P2-P1-PDROP3*SIGN(1.0, Q3))/II3
Q3=INTEG(Q3DOT,0.) $ Q3P=Q3*1.585E4
QV3=INTEG(Q3,0.)

QP=Q1P+Q2P
V=QV1+QV2+QV3
V1=(-AP*(X-XKEEP)+V)*(12./0.3048)**3+V10
V2=-V*(12./0.3048)**3+V20
V AIR=V A10+AP*(X-XKEEP)-V-V10*(0.3048/12.)**3.

FT=KR*X+BR*XDOT+AP*(P0-P1)

END $ "OF DERIVATIVE"

DISCRETE MOUNT
PROCEDURAL

" DECOUPLER GAP DYNAMICS"

" PRES=.TRUE.: VACUUM PRESSURE STAGE"
" PRES=.FALSE.: COMPRESSIVE PRESSURE STAGE"
" CPLD=.TRUE.: THE DECOUPLER HAS BEEN COUPLED"
" CPLD=.FALSE.: THE DECOUPLER IS MOVING"

IF(CPLD) GO TO 200
" DECOUPLER IS COUPLED"
GAP=VGAP $ CPLD=.TRUE. $ PRES=.NOT.PRES
LLCD3=LLCODEC$ LI3=LENGTH(LLCD3,ACODEC)*1.E-3
DI3=DI(A(ACODEC)+1.E-3
A13=PHI*DI3**2./4.
XKEEP=X $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
GO TO 300

200. CONTINUE
IF (DGAP,LT,0.1) GO TO 300
" DECOUPLER IS DECOUPLED"
CPLD=FALSE.
LLCD3=LCODED $ Li3=LENGTH(LLCD3,ACODED)*1,E-3
Di3=DI(A,ACODED)*1,E-3
A3=PHI*DI3**2./4.$
XKEEP=X $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0. $ Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
300.. CONTINUE
CALL LOGD(.TRUE.)
END $ "OF PROCEDURAL"
END $ "OF DISCRETE"

WRITE (50,40) T
WRITE (51,40) P1
WRITE (52,40) P2
IF (T,GE,PERIOD) GO TO 80
WRITE(2,20) T,X,P1,P2,V1,V2
20.. FORMAT(F8.5,F8.5,2E18.10,6E11.3)
80.. CONTINUE
IF (T,L.T.(TS2-0.001*PERIOD)) GO TO 100
IF (T,G.T.TF1) GO TO 90
TT=T-TS2 $ XX=X*1000. $ XXDOT=XdOT*1000. $ PD=P2-P1
WRITE(20,40) TT
WRITE(21,40) XX
WRITE(22,40) FT
WRITE(23,40) P1
WRITE(24,40) P2
WRITE(25,40) Q1P
WRITE(26,40) Q2P
WRITE(30,40) Q3P
WRITE(27,40) XXDOT
WRITE(28,40) PD
WRITE(29,40) V2
90.. CONTINUE
IF (T,L.T.TS1) GO TO 100
WRITE (10,40) T
WRITE (11,40) X
WRITE (12,40) FT
40.. FORMAT (E15.5)
100.. CONTINUE
TERMT(T,GE,TF)
END
END

SUBROUTINE ASS

OPEN UNIT=2, FILE="PRC.LOG", STATUS="UNKNOWN"
CLOSE (UNIT=2, STATUS="DELETE")
OPEN (UNIT=2, FILE="PRC.LOG", STATUS="NEW,FORM="FORMATTED")

WRITE(2,310)
310.. FORMAT( ' TIME X VAIL... PIE V1 V Q'),
OPEN (UNIT=10, FILE='[.TMP]T1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=10, STATUS='DELETE')
OPEN (UNIT=10, FILE='[.TMP]T1.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=11, FILE='[.TMP]X1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=11, STATUS='DELETE')
OPEN (UNIT=11, FILE='[.TMP]X1.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=12, FILE='[.TMP]FT1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=12, STATUS='DELETE')
OPEN (UNIT=12, FILE='[.TMP]FT1.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=20, FILE='[.TMP]T.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=20, STATUS='DELETE')
OPEN (UNIT=20, FILE='[.TMP]T.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=21, FILE='[.TMP]X.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=21, STATUS='DELETE')
OPEN (UNIT=21, FILE='[.TMP]X.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=22, FILE='[.TMP]FT.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=22, STATUS='DELETE')
OPEN (UNIT=22, FILE='[.TMP]FT.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=23, FILE='[.TMP]P1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=23, STATUS='DELETE')
OPEN (UNIT=23, FILE='[.TMP]P1.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=24, FILE='[.TMP]P2.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=24, STATUS='DELETE')
OPEN (UNIT=24, FILE='[.TMP]P2.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=25, FILE='[.TMP]Q1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=25, STATUS='DELETE')
OPEN (UNIT=25, FILE='[.TMP]Q1.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=26, FILE='[.TMP]Q2.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=26, STATUS='DELETE')
OPEN (UNIT=26, FILE='[.TMP]Q2.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=27, FILE='[.TMP]XDOT.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=27, STATUS='DELETE')
OPEN (UNIT=27, FILE='[.TMP]XDOT.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=28, FILE='[.TMP]PD.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=28, STATUS='DELETE')
OPEN (UNIT=28, FILE='[.TMP]PD.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=29, FILE='[.TMP]V2.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=29, STATUS='DELETE')
OPEN (UNIT=29, FILE='[.TMP]V2.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=30, FILE='[.TMP]QD.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=30, STATUS='DELETE')
OPEN (UNIT=30, FILE='[.TMP]QD.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=50, FILE='[.TMP]TT.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=50, STATUS='DELETE')
OPEN (UNIT=50, FILE='[.TMP]TT.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=51, FILE='[.TMP]PIT.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=51, STATUS='DELETE')
OPEN (UNIT=51, FILE='[.TMP]PIT.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=52, FILE='[.TMP]P2T.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=52, STATUS='DELETE')
OPEN (UNIT=52, FILE='[.TMP]P2T.DAT', STATUS='NEW', FORM='FORMATTED')

RETURN
END
PROGRAM FREQUENCY DOMAIN RESPONSE OF REGULAR HYDRAULIC MOUNT

DESCRIPTION
This program simulates the frequency domain response of the regular hydraulic mount in the environment of component testing. Two inertia track paths and decoupler are modeled. Upward motion is taken positive. If DGAP < 0.1 mm, the mount is taken as the inertia track mount. Frequency responses are calculated with reference to the last response cycles after the steady state has been reached for each excitation frequency. As described in Section 3.5.5, the Fourier filter algorithm is implemented to extract the mean value and harmonic amplitudes (up to 3rd) from a nonsinusoidal transmitted force. Only fundamental harmonic amplitude is extracted for Q1 and Q2. On the other hand, the maximum and minimum values are found for the upper chamber pressure.

The following frequency responses can be obtained from this program.

- ANG1 : loss angle (deg)
- ANG2 : phase lead of second harmonic transmitted force w.r.t X (deg)
- ANG3 : phase lead of third harmonic transmitted force w.r.t X (deg)
- AQ1P : phase lead of Q1 with reference to X (deg)
- AQ2P : phase lead of Q2 with reference to X (deg)
- KDN : dynamic stiffness modulus (N/mm)
- MAQ0 : mean value of transmitted force (N)
- MAQ1 : amplitude of fundamental harmonic of transmitted force (N)
- MAQ2 : amplitude of second harmonic of transmitted force (N)
- MAQ3 : amplitude of third harmonic of transmitted force (N)
- MQ1P : amplitude of fundamental harmonic of Q1 (gpm)
- MQ2P : amplitude of fundamental harmonic of Q2 (gpm)
- MXP1 : maximum pressure of upper chamber (KPa)
- MNP1 : minimum pressure of upper chamber (KPa)

LIST OF VARIABLES AND PARAMETERS
- HZMN : minimum frequency in the frequency range of concern (Hz)
- HZMX : maximum frequency in the frequency range of concern (Hz)
- INP : coincident component of transmitted force
- INQ1P : coincident component of Q1
- INQ2P : coincident component of Q2
- QAD : quadrature component of transmitted force
- QUQ1P : quadrature component of Q1
- QUQ2P : quadrature component of Q2

INITIAL
RESET('NOEVAL')
" OPEN DATA LOG FILES "
CALL ASS
" USE NCYCLE=3, FOR AMP>.1; NCYCLE=7, FOR AMP=.01"
" LCODE1=1 FOR CHANNEL CODE A "
" LCODE1=14 FOR CHANNEL CODE N "
" LCODE1=18 FOR DUAL CHANNEL DEVICE "

LOGICAL CPLD, PRES
INTEGER LCODE1, LCODED, LCODEC
CONSTANT PHI=3.141592, RHO=1059., MU=3.17E-3
CONSTANT DP=.08, GAM=1.0, LAIR=0.0005
CONSTANT AD=.0023, DGAP=0.7
CONSTANT NCYCLE=3., AMP=1.0
CONSTANT PRE=1200., PATM=101300.
CONSTANT LCODE1=14, AICODE1=2.0
CONSTANT LCODED=19, AICODED=5., LCODEC=19, ACODEC=6.
CONSTANT HZMN=1., HZMX=50.

TABLE KRBL.1.22 ...
   /1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11,...
   12., 13., 14., 15., 20., 25., 30., 35., 40., 45., 50., ...
   277.,278.,280.,281.,282.,283.,283.,284.,284.,...

TABLE BRTBL.1.22 ...
   /1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11,...
   12., 13., 14., 15., 20., 25., 30., 35., 40., 45., 50., ...
   149.,140.,132.,125.,98.,88.5,79.9,74.69.3,67.64.5/   

" INITIAL CONDITIONS; PIBAR(PA), V2BAR(IN**3), V1BAR(CC)"
" V1BAR IS TRANSFORMED TO IN**3 LATER"

TABLE PBAR.1.19 ...
   /200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
   1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
   1900., 2000., ...
   101590.7, 102086.1, 102867.7, 103917.7, 105236.7, ...
   106762.8, 108437.3, 110236.0, 112137.2, 114136.2, ...
   116405.6, 119129.9, 122604.5, 127144.8, 133094.8, ...
   140218.9, 148879.0, 158735.6, 169738.0/

TABLE V1BAR.1.19 ...
   /200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
   1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
   1900., 2000., ...
   .1038., .1704., .2482., .3333., .425., .5192., .6133., .7069., ...
   .7986., .8922., .992., 1.1062., 1.2448., 1.4164., 1.6289., ...
   1.8693, 2.1458, 2.4445, 2.7624/

TABLE V2BARI.1.19 ...
   /200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
   1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
   1900., 2000., ...
   .304., .454., .602., .746., .89., 1.032., 1.172, 1.312., 1.45., ...
   1.588., 1.724., 1.856, 1.982, 2.1, 2.21., 2.306, 2.394, ...
   2.472, 2.542/
"CHANNEL CODE= A B C D E F G H I J K L M N O P Q DUAL DECPL"
"LCODE= 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19"
"MM UNIT"

```
TABLE LENGTH.2,19.8 ...
/1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11., 12., 13., ...
14., 15., 16., 17., 18., 19., ...
0., 1., 2., 2.5, 3., 4., 5., 6., ...
25.9, 36.7, 47.5, 58.2, 69.0, 79.8, 90.6, 101.4, 112.2, ...
123.0, 133.7, 144.5, 155.3, 166.1, 176.9, 187.7, 198.5, ...
166.1, 5., ...
25.5, 36.1, 46.8, 57.4, 68.0, 78.7, 89.3, 99.9, 110.5, ...
121.2, 131.8, 142.4, 153.1, 163.7, 174.3, 184.9, 195.6, ...
163.7, 5., ...
25.1, 35.6, 46.1, 56.5, 67.0, 77.5, 88.0, 98.4, 108.9, ...
119.4, 129.9, 140.3, 150.8, 161.3, 171.7, 182.2, 192.7, ...
161.3, 5., ...
24.9, 35.3, 45.7, 56.1, 66.5, 76.9, 87.3, 97.7, 108.1, ...
118.5, 128.9, 139.3, 149.7, 160.1, 170.5, 180.8, 191.2, ...
160.1, 5., ...
24.8, 35.1, 45.4, 55.7, 66.0, 76.3, 86.6, 97.0, 107.3, ...
117.6, 127.9, 138.2, 148.5, 158.8, 169.2, 179.5, 189.8, ...
158.8, 5., ...
24.3, 34.4, 44.5, 54.6, 64.7, 74.8, 84.9, 95.0, 105.1, ...
115.2, 125.3, 135.4, 145.5, 155.6, 165.7, 175.8, 185.9, ...
155.6, 5., ...
5., 5., 5., 5., 5., 5., 5., 5., 5., 5., ...
5., 5., 5., 5., 5., 5., 7., ...
6., 6., 6., 6., 6., 6., 6., 6., 6., 6., ...
6., 6., 6., 6., 6., 7./
```

"AREA REDUCER= FULL 1.0 2.0 2.5 3.0 4.0 5.0 6.0"
"ACODE=. 0. 1.0 2.0 2.5 3.0 4.0 5.0 6.0 MM**2 UNIT"

```
TABLE DIA.1,8 ...
/0., 1.0, 2.0, 2.5, 3.0, 4.0, 5.0, 6.0, ...
7.501, 6.746, 5.891, 5.409, 4.906, 3.772, 30., 0.1/
```

```
TABLE TPFAC.1,9 ...
/1., 2., 3., 4., 5., 6., 7., 8., 50,...
500., 200., 200., 100., 100., 50., 50., 50., 50./

AP=PHI*DP*DP/4,
XAMP=AMP*0.001
VGAP=AD*DGAP*0.001
```

"VAIR0 = INITIAL GAS VOLUME; JUGGLE WITH LAIR TO CHANGE VAIR0"

```
VAIR0=LAIR*AP
```

"GEOMETRIES FOR TWO INERTIA TRACKS"
LLCD1=LCODE1 $ LI1=LENGTH(LLCD1,ACODE1)*1.0-3 $ LI2=0.2115-LI1
D11=DIA(ACODE1)*1.0-3 $ D12=D11
AI1=PHI*D11**2./4. $ AI2=AI1

MAG=0. $ ANG=0. $ KDN=0.
MQ1P=0. $ AQ1P=0. $ MQ2P=0. $ AQ2P=0.
HZ=HZMN

400.      CONTINUE
P0=PBAR(PRE) $ V20=V2BAR(PRE) $ V10=V1BAR(PRE)/2.54**3.

" DECOUPLER GEOMETRY"
LLC3=LCODED $ L13=LENGTH(LLC3,ACODED)*1.E-3
D13=DIA(ACODED)*1.E-3
Al3=PHI*DI3**2./4.

CPLD=.FALSE. $ PRES=.FALSE.

KR=KRTBL(HZ)*1.E3
BR=BRTBL(HZ)
MXP1=0. $ MNP1=1000.
INPH=0. $ QUAD=0.
INQ1P=0. $ QUAD1P=0. $ INQ2P=0. $ QUAD2P=0.

V1=V10 $ V2=V20
Q1=0. $ Q2=0. $ Q3=0. $ P1=P0 $ P2=P0
PFAC=TFAC(HZ)
OMEGA=HZ**2.*PHI
X=0. $ XKEEP=0. $ GAP=VGAP/2.
PERIOD=1./HZ $ TF=NCYCLE*PERIOD
TS1=TF-PERIOD $ TS2=TF-PERIOD*2.
CINT=PERIOD/PFAC

END

DYNAMIC

DERIVATIVE

SELP=RSW(PRES, P1-P2-1000., P2-P1-1000.)
SELD=RSW(CPLD, SELP, ABS(QV3)-GAP)
SCHEDULE MOUNT.XP.SELD

PROCEDURAL (P1,P2,P1E,P2E,P1EG,P2EG,PDROP1,PDROP2=V1,V2,Q1,Q2,VAIR)

P2EG=0.83420*V2**2.5-0.25117*V2**6.+0.159415*V2**6.5
P2=P2EG*6894.41+PATM
IF(P2.LE.0.) P2=0.
P2E=P2/6894.41
IF (V1.LE.0.) GO TO 100
P1EG=-76.86729*V1+180.81844*V1**2.*V1**2.*7./6.+19.46422*V1**2.5
P1=P1EG*6894.41+PATM
IF(P1.LE.0.) P1=0.
GO TO 110

100.      CONTINUE
P1=PATM*(VAIR(0./VAIR)**GAM

110.      CONTINUE
P1E=P1/6894.41

" PRESSURE DROP ASSOCIATED WITH TURBULENT FLOW THROUGH CHANNELS"

GO TO (1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18), LCODE1
GO TO 20
1.. CONTINUE  
   PDROP1=0.634*1.732E12*ABS(Q1)**2.  
   GO TO 20  
2.. CONTINUE  
   GO TO 20  
3.. CONTINUE  
   GO TO 20  
4.. CONTINUE  
   GO TO 20  
5.. CONTINUE  
   GO TO 20  
6.. CONTINUE  
   GO TO 20  
7.. CONTINUE  
   GO TO 20  
8.. CONTINUE  
   GO TO 20  
9.. CONTINUE  
   GO TO 20  
10.. CONTINUE  
    GO TO 20  
11.. CONTINUE  
    GO TO 20  
12.. CONTINUE  
    GO TO 20  
13.. CONTINUE  
    GO TO 20  
14.. CONTINUE  
   PDROP1=1.10*1.732E12*ABS(Q1)**2.  
   PDROP2=1.99*1.732E12*ABS(Q2)**2.  
   GO TO 20  
15.. CONTINUE  
   PDROP1=1.10*1.732E12*ABS(Q1)**2.  
   PDROP2=1.99*1.732E12*ABS(Q2)**2.  
   GO TO 20  
16.. CONTINUE  
   PDROP1=1.10*1.732E12*ABS(Q1)**2.  
   PDROP2=1.99*1.732E12*ABS(Q2)**2.  
   GO TO 20  
17.. CONTINUE  
   GO TO 20  
18.. CONTINUE  
   PDROP1=0.634*1.732E12*ABS(Q2)**2.  
   PDROP2=0.634*1.732E12*ABS(Q2)**2.  
20.. CONTINUE  
   PDROP3=(Q3/0.6/AI3)**2.*RHO/2.  

END  

X=XAMP*SIN(OMEGA*T)  
XDOT=XAMP*OMEGA*COS(OMEGA*T)  

" FLOW THROUGH INERTIA TRACK PATH I"  
REH1=DI1/MU*SQR(T(2.*PDROP1*RHO)  
II1=RHO*(LI1/11)  
Q1DOT=(P2-P1-PDRP1*SIGN(1.0, Q1)/II1
Q1=INTEG(Q1DOT,0.) $ Q1P=Q1*1.585E4
QV1=INTEG(Q1,0.)

" FLOW THROUGH INERTIA TRACK PATH 2"
REH2=DI2/MU*SQR(2.*PDROP2*RHO)
II2=RHO*(LI2)/AI2
Q2DOT=(P2-P1-PDROP2*SIGN(1.0, Q2))/II2
Q2=INTEG(Q2DOT,0.) $ Q2P=Q2*1.585E4
QV2=INTEG(Q2,0.)

" FLOW THROUGH DECOUPLER"
RE3=4.*RHO*ABS(Q3)/PHID/II3/MU
II3=RHO*(LI3)/AI3
Q3DOT=(P3-P3-PDROP3*SIGN(1.0, Q3))/II3
Q3=INTEG(Q3DOT,0.) $ Q3P=Q3*1.585E4
QV3=INTEG(Q3,0.)
QP=Q1P+Q2P
V=QV1+QV2+QV3

V1=(-AP*(X-XKEEP)+V)*(12./0.3048)**3+V10
V2=V*(12./0.3048)**3+V20
VAIR=VAIR0+AP*(X-XKEEP)-V-V10*(0.3048/12.)**3.
FT=KR*X+BR*XDOT+AP*(P0-P1)

" TIME HISTORY PROCESSING"
OMXP1=RSW(T,GT,TS1, P1, 0.)
OMNP1=RSW(T,GT,TS1, P1, 1.E10)
MXP1=AMAX1(MXP1, OMXP1*0.001)
MNP1=AMIN1(MNP1, OMNP1*0.001)

" FREQUENCY RESPONSE PROCESSING"
OUTSIG=RSW(T,GE,TS1, FT, 0.)
INPH1=INTEG(OUTSIG*SIN(OMEGA*T),0.)
QUAD1=INTEG(OUTSIG*COS(OMEGA*T),0.)
INPH2=INTEG(OUTSIG*SIN(2.*OMEGA*T),0.)
QUAD2=INTEG(OUTSIG*COS(2.*OMEGA*T),0.)
INPH3=INTEG(OUTSIG*SIN(3.*OMEGA*T),0.)
QUAD3=INTEG(OUTSIG*COS(3.*OMEGA*T),0.)
MAG0=INTEG(OUTSIG,0.)
OUTQ1P=RSW(T,GE,TS1, Q1P, 0.)
INPQ1P=INTEG(OUTQ1P*SIN(OMEGA*T),0.)
QUDQ1P=INTEG(OUTQ1P*COS(OMEGA*T),0.)
OUTQ2P=RSW(T,GE,TS1, Q2P, 0.)
INPQ2P=INTEG(OUTQ2P*SIN(OMEGA*T),0.)
QUDQ2P=INTEG(OUTQ2P*COS(OMEGA*T),0.)

END $ "OF DERIVATIVE"

DISCRETE MOUNT
PROCEDURAL
" DECOUPLER GAP DYNAMICS"

" PRES=.TRUE.: VACUUM PRESSURE STAGE"
" PRES=.FALSE.: COMPRESSIVE PRESSURE STAGE"
" CPLD=.TRUE.: THE DECOUPLER HAS BEEN COUPLED"
" CPLD=.FALSE.: THE DECOUPLER IS MOVING"

IF(CPLD) GO TO 200
" DECOUPLER IS COUPLED"
    GAP=VGAP $ CPLD=.TRUE. $ PRES=.NOT.PRES
    LLCD3=LCodec $ L13=LENGTH(LLCD3,ACODEC)*1.E-3
    DI3=DI(A,ACODEC)*1.E-3
    A13=PHI(DI3)**2/4.
    XKEEP=X $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
    Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
    GO TO 300
200.. CONTINUE
    IF (DGAP.LT.0.1) GO TO 300
" DECOUPLER IS DECOUPLED"
    CPLD=.FALSE.
    LLCD3=LCodec $ L13=LENGTH(LLCD3,ACODED)*1.E-3
    DI3=DI(A,ACODED)*1.E-3
    A13=PHI(DI3)**2/4.
    XKEEP=X $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
    Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
300.. CONTINUE
    END $ "OF PROCEDURAL"
    END $ "OF DISCRETE"

TERMT(T,G,E,T)
END $ "END OF DYNAMIC"

TERMINAL

MAGO=MAG0/PERIOD
INP1=INPH1/PERIOD $ QAD1=QUAD1/PERIOD
MAG1=SQRT(INP1**2.+QAD1**2.)*2. $ ANG1=ATAN2(QAD1,INP1)*180./PHI
KDN=MAG1/AMP
INP2=INPH2/PERIOD $ QAD2=QUAD2/PERIOD
MAG2=SQRT(INP2**2.+QAD2**2.)*2. $ ANG2=ATAN2(QAD2,INP2)*180./PHI
INP3=INPH3/PERIOD $ QAD3=QUAD3/PERIOD
MAG3=SQRT(INP3**2.+QAD3**2.)*2. $ ANG3=ATAN2(QAD3,INP3)*180./PHI

INQ1P=INQP1P/PERIOD $ QUQ1P=QUDQ1P/PERIOD
MQ1P=SQRT(INQ1P**2.+QUQ1P**2.)*2.
AQ1P=ATAN2(QUQ1P,INQP1P)*180./PHI

INQ2P=INQP2P/PERIOD $ QUQ2P=QUDQ2P/PERIOD
MQ2P=SQRT(INQ2P**2.+QUQ2P**2.)*2.
AQ2P=ATAN2(QUQ2P,INQP2P)*180./PHI

WRITE (2,20) HZ,MAG1,ANG1,MQ1P,AQ1P,MQ2P,AQ2P
20.. FORMAT (7E1.3)
   WRITE (3,30) HZ,MAG0,MAG1,MAG2,MAG3,ANG1,ANG2,ANG3
30.. FORMAT (8E11.3)
   WRITE (10,40) HZ
   WRITE (11,40) MAG
WRITE (12,40) ANG
WRITE (13,40) MQ1P
WRITE (14,40) AQ1P
WRITE (17,40) MQ2P
WRITE (18,40) AQ2P
WRITE (19,40) MXP1
WRITE (20,40) MNP1

40. FORMAT (E15.5)
   " CALL LOGD(.TRUE.)"
   DELHZ=RSW(HZ.LT.10.,3.,5.0)
   HZ=HZ+DELHZ
   IF (HZ.LE.HZMX) GO TO 400
END $ "OF TERMINAL"

END $ "OF PROGRAM"

SUBROUTINE ASS

OPEN (UNIT=2, FILE='PRC.LOG',STATUS='UNKNOWN')
CLOSE (UNIT=2, STATUS='DELETE')
OPEN (UNIT=2, FILE='PRC.LOG', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=3, FILE='HAR.LOG', STATUS='UNKNOWN')
CLOSE (UNIT=3, STATUS='DELETE')
OPEN (UNIT=3, FILE='HAR.LOG', STATUS='NEW', FORM='FORMATTED')

WRITE(2,310)
   FORMAT(10, $ 'HZ MAG ANG ',',
                   1'MQ1P AQ1P MQ2P AQ2P MAG0')
WRITE(3,315)
   FORMAT(10, $ 'HZ MAG0 MAG1 ',',
                   1'MAG2 MAG3 ANG1 ANG2 ANG3')

OPEN (UNIT=10, FILE='[.TMP]HZ.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=10, STATUS='DELETE')
OPEN (UNIT=10, FILE='[.TMP]HZ.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=11, FILE='[.TMP]MAG.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=11, STATUS='DELETE')
OPEN (UNIT=11, FILE='[.TMP]MAG.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=12, FILE='[.TMP]ANG.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=12, STATUS='DELETE')
OPEN (UNIT=12, FILE='[.TMP]ANG.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=13, FILE='[.TMP]MQ1P.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=13, STATUS='DELETE')
OPEN (UNIT=13, FILE='[.TMP]MQ1P.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=14, FILE='[.TMP]AQ1P.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=14, STATUS='DELETE')
OPEN (UNIT=14, FILE='[.TMP]AQ1P.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=17, FILE='[.TMP]MQ2P.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=17, STATUS='DELETE')
OPEN (UNIT=17, FILE='[.TMP]MQ2P.DAT', STATUS='NEW', FORM='FORMATTED')
OPEN (UNIT=18, FILE='.TMP\AQ2P.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=18, STATUS='DELETE')
OPEN (UNIT=18, FILE='.TMP\AQ2P.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=19, FILE='.TMP\MXP1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=19, STATUS='DELETE')
OPEN (UNIT=19, FILE='.TMP\MXP1.DAT', STATUS='NEW', FORM='FORMATTED')

OPEN (UNIT=20, FILE='.TMP\MNP1.DAT', STATUS='UNKNOWN')
CLOSE (UNIT=20, STATUS='DELETE')
OPEN (UNIT=20, FILE='.TMP\MNP1.DAT', STATUS='NEW', FORM='FORMATTED')

RETURN
END
PROGRAM HARMONIC RESPONSE IN TIME DOMAIN OF VEHICLE MODEL

DESCRIPTION
This program simulates the time domain response of the vehicle model installed with passive hydraulic mount. Two harmonic excitations are possible through the engine force or road profile acceleration. The initial conditions of the hydraulic mount is related with static equilibrium under the engine mass ME. Upward motion is taken positive.
If DGAP<0.1 mm, the mount is taken as the inertia track mount.

LIST OF VARIABLES AND PARAMETERS
AMP: amplitude of road profile acceleration (m/s²)
BS: damping coefficient of suspension system (N·s/m)
KS: elastic stiffness of suspension system (N/m)
ME: engine mass (Kg)
MS: sprung mass (Kg)
XD: relative engine displacement (mm)
XE: displacement of engine mass (m)
XEDOT: velocity of engine mass (m/s)
XEDOT2: acceleration of engine mass (m/s²)
XS: displacement of sprung mass (m)
XS DOT: velocity of sprung mass (m/s)
XS DOT2: acceleration of sprung mass (m/s²)
XU: road profile displacement (m)
XUDOT: road profile velocity (m/s)
XUDOT2: road profile acceleration (m/s²)

INITIAL
OPEN DATA LOG FILES
CALL ASS
LOGICAL CPLD, PRES
INTEGER LCODE1, LCODED, LCODEC
CONSTANT PHI=3.141592, RHO=1059, MU=3.17E-3
CONSTANT DP=.08, GAM=1.0, LAIR=0.0005, PATM=101300.
CONSTANT AD=.0023, DGAP=0.7
CONSTANT AMP=0., FAMP=100.
CONSTANT HZ=20., NCYCLE=5., PFAC=50.
CONSTANT LCODE1=14, ACODE1=2.0
CONSTANT LCODED=19, ACODED=5., LCODEC=19, ACODEC=6.
CONSTANT ME=122.3, KS=2.64, BS=1400., MS=270.

**TABLE KRTBL.1,22**

| 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 20, 25, 30, 35, 40, 45, 50, 277, 278, 280, 281, 282, 283, 284, 285, 286, 287, 288, 289, 290, 291, 292, 294 |

**TABLE BRTBL.1,22**

| 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 20, 25, 30, 35, 40, 45, 50, 1540, 779, 521, 396, 322, 273, 237, 211, 191, 174, 160, 149, 140, 132, 125, 98, 88.5, 79.9, 74, 69.3, 67, 64.5 |

"INITIAL CONDITIONS: P1BAR(PA), V2BAR(IN**3), V1BAR(CC)"

"V1BAR IS TRANSFORMED TO IN**3 LATER"

**TABLE P1BAR.1,19**

| 200., 300., 400., 500., 600., 700., 800., 900., 1000., 1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., 1900., 2000., 101590.7, 102086.1, 102867.7, 103917.7, 105236.7, 106762.8, 108437.3, 110236.0, 112113.7, 114136.2, 116405.6, 119129.9, 122604.5, 127144.8, 133094.8, 140218.9, 148879.0, 158735.6, 169738.0 |

**TABLE V1BAR.1,19**


**TABLE V2BAR.1,19**


"CHANNEL CODE= A B C D E F G H I J K L M N O P Q DUAL DECPL"

"LCODE= 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19"

"MM UNIT"

**TABLE LENGTH.2,19,8**

| 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14., 15., 16., 17., 18., 19., 0., 1., 2., 2.5, 3., 4., 5., 6., 25.9, 36.7, 47.5, 58.2, 69.0, 79.8, 90.6, 101.4, 112.2, 123.0, 133.7, 144.5, 155.3, 166.1, 176.9, 187.7, 198.5, 166.1, 5., 25.5, 36.1, 46.8, 57.4, 68.0, 78.7, 89.3, 99.9, 110.5, 121.2, 131.8, 142.4, 153.1, 163.7, 174.3, 184.9, 195.6, 163.7, 5., 25.1, 35.6, 46.1, 56.5, 67.0, 77.5, 88.0, 98.4, 108.9 |
119.4, 129.9, 140.3, 150.8, 161.3, 171.7, 182.2, 192.7, ...
161.3, 5., ...
24.9, 35.3, 45.7, 56.1, 66.5, 76.9, 87.3, 97.7, 108.1, ...
118.5, 128.9, 139.3, 149.7, 160.1, 170.5, 180.8, 191.2, ...
160.1, 5., ...
24.8, 35.1, 45.4, 55.7, 66.0, 76.3, 86.6, 97.0, 107.3, ...
117.6, 127.9, 138.2, 148.5, 158.8, 169.2, 179.5, 189.8, ...
158.8, 5., ...
24.3, 34.4, 44.5, 54.6, 64.7, 74.8, 84.9, 95.0, 105.1, ...
115.2, 125.3, 135.4, 145.5, 155.6, 165.7, 175.8, 185.9, ...
155.6, 5., ...
5., 5., 5., 5., 5., 5., 5., 5., 5., 5., 5., ...
5., 5., 5., 5., 5., 7., ...
6., 6., 6., 6., 6., 6., 6., 6., 6., 6., 6., ...
6., 6., 6., 6., 6., 6., 7. /

" AREA REDUCER = FULL 1.0 2.0 2.5 3.0 4.0 5.0 6.0"
" ACODE = 0. 1.0 2.0 2.5 3.0 4.0 5.0 6.0 MM**2 UNIT"

" TABLE DIA.18 ... ".
" /0., 1.0, 2.0, 2.5, 3.0, 4.0, 5.0, 6.0, ...
" 7.501, 6.746, 5.891, 5.409, 4.906, 3.772, 30., 0.1/

" TABLE TPFA1.18 ... ".
" /3., 4., 5., 6., 7., 8., 9., 10., 11., 12., 13., ...
" 14., 15., 16., 17., 18., 19., 20., ...
" 200., 200., 100., 100., 50., 50., 50., 50., 50., 50., ...
" 50., 50., 50., 50., 50., 50., 50., 50., 50./

AP=PHI*DP*DP/4.
PRE=ME**9.81
KR=KRTBL(HZ)*1.E3
BR=BRTBL(HZ)
PO=PIBAR(PRE) $ V20=V2BAR(PRE) $ V10=V1BAR(PRE)/2.54**3.

" GEOMETRIES FOR TWO INERTIA TRACKS"
LLCD1=LCODE1 $ L1=LENGTH(LLCD1,ACODE1)*1.E-3 $ L12=0.2115-L11
D11=D1A(ACODE1)*1.E-3 $ D12=D11
A11=PHI*D11**2./4. $ A12=A11

" DECOUPLER GEOMETRY"
LLCD3=LCODE3 $ L13=LENGTH(LLCD3,ACODE3)*1.E-3
D13=D1A(ACODE3)*1.E-3
A13=PHI*D13**2./4.

" VAIR0 = INITIAL GAS VOLUME; JUGGLE WITH LAIR TO CHANGE VAIR0"

VAIR0=LAIR*AP
V1=V10 $ V2=V20
Q1=0. $ Q2=0. $ Q3=0. $ PI=P0 $ P2=P0
OMEGA=HZ*2.*PHI
CPLD=FALSE. $ PRES=FALSE.
VGAP=AD*DGAP*0.001 $ XDKEEP=0. $ GAP=VGAP/2.

" TPFA1=TPFA1(HZ)"
PERIOD=1./HZ $ CINT=PERIOD/PFAC
TF1=NCYCLE*PERIOD-PERIOD*0.25
TF2=NCYCLE*PERIOD
TF=NCYCLE*PERIOD-CINT
TS1=TF2-PERIOD $ TS2=TF2-PERIOD*2.

XE=0. $ XEDOT=0. $ XS=0. $ XSDOT=0. 
"FAMP=ECEN*OMEGA**2."

END $ "OF INITIAL"

DYNAMIC

DERIVATIVE

SELP=RSW(PRES, P1-P2-100., P2-P1-100.)
SELD=RSW(CPLD, SELP, ABS(QV3)-GAP)
SCHEDULE MOUNT.XP.SELD

PROCEDURAL (P1,P2,P1E,P2E,P1EG,P2EG,PDROP1,PDROP2=V1,V2,Q1,Q2,VAIR)

P2EG=0.83420*V2**2.5-0.25117*V2**6.+0.159415*V2**6.5
P2=P2EG*6894.41+PATM
IF(P2.LE.0.) P2=0.
P2E=P2/6894.41
IF (V1.LE.0.) GO TO 100
P1EG=-76.86729*V1+180.81844*V1**(7./6.)+19.46422*V1**2.5
P1=P1EG*6894.41+PATM
IF(P1.LE.0.) P1=0.
GO TO 110

100.. CONTINUE
P1=PATM*(VAIR0/VAIR)**GAM

110.. CONTINUE
P1E=P1/6894.41

" PRESSURE DROP ASSOCIATED WITH TURBULENT FLOW THROUGH CHANNELS"

GO TO (1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18), LCODE1
GO TO 20

1.. CONTINUE
PDROP1=0.634*1.732E12*ABS(Q1)**2.
GO TO 20

2.. CONTINUE
GO TO 20

3.. CONTINUE
GO TO 20

4.. CONTINUE
GO TO 20

5.. CONTINUE
GO TO 20

6.. CONTINUE
GO TO 20

7.. CONTINUE
GO TO 20

8.. CONTINUE
GO TO 20

9.. CONTINUE
GO TO 20

10.. CONTINUE
GO TO 20
11. CONTINUE
   GO TO 20
12. CONTINUE
   GO TO 20
13. CONTINUE
   GO TO 20
14. CONTINUE
   PDROP1=1.10*1.732E12*ABS(Q1)**2.
   PDROP2=1.99*1.732E12*ABS(Q2)**2.
   GO TO 20
15. CONTINUE
   PDROP1=1.10*1.732E12*ABS(Q1)**2.
   PDROP2=1.99*1.732E12*ABS(Q2)**2.
   GO TO 20
16. CONTINUE
   PDROP1=1.10*1.732E12*ABS(Q1)**2.
   PDROP2=1.99*1.732E12*ABS(Q2)**2.
   GO TO 20
17. CONTINUE
   GO TO 20
18. CONTINUE
   PDROP1=1.10*1.732E12*ABS(Q1)**2.
   PDROP2=0.634*1.732E12*ABS(Q2)**2.
20. CONTINUE
   PDROP3=(Q3/0.6/A13)**2.*RHO/2.

END

F=FAMP*SIN(OMEGA*T)
XUDOT2=AMP*SIN(OMEGA*T)
XUDOT=AMP/OMEGA*COS(OMEGA*T)
XU=-AMP/(OMEGA**2)*SIN(OMEGA*T)

XSDOT2=(-AP*(P1-P0)+KS*(XU-XS)+BS*(XUDOT-XSDOT))
   +KR*(XE-XS)+BR*(XEDOT-XSDOT))/MS
XSDOT=INTEG(XSDOT2,0.)
XS=INTEG(XSDOT,0.)

XEDOT2=(-KR*(XE-XS)-BR*(XEDOT-XSDOT)+AP*(P1-P0))/ME
XEDOT=INTEG(XEDOT2,0.)
XE=INTEG(XEDOT,0.)

XD=(XE-XS)*1000.

" FLOW THROUGH INERTIA TRACK PATH 1"
REH1=(D11/MU*SQR(T2.*PDROP1*RHO))
II1=RHO*(L11)/AI1
Q1DOT=(P2-P1-PDROP1*SIGN(1.0, Q1))/II1
Q1=INTEG(Q1DOT,0.) $ Q1P=Q1*1.585E4
QV1=INTEG(Q1,0.)

" FLOW THROUGH INERTIA TRACK PATH 2"
REH2=(D12/MU*SQR(T2.*PDROP2*RHO))
II2=RHO*(L12)/AI2
Q2DOT=(P2-P1-PDROP2*SIGN(1.0, Q2))/II2
Q2=INTEG(Q2DOT,0.) $ Q2P=Q2*1.585E4
QV2=INTEG(Q2,0.)
" FLOW THROUGH DECOUPLER"
RE3=4.*RHO*ABS(Q3)/PHI/DI3/MU
II3=RHO*(LI3/AIL3
Q3DOT=(P2-P1-PDROP3*SIGN(1.0,Q3))/II3
Q3=INTEG(Q3DOT,0.) $ Q3P=Q3*1.585E4
QV3=INTEG(Q3,0.)
QP=Q1P+Q2P
V=QV1+QV2+QV3
V1=(-AP*(XE-XS-XDKEEP)+V)*(12./0.3048)**3+V10
V2=V*(12./0.3048)**3+V20
VAIR=VAIR0+AP*(XE-XS-XDKEEP)-V-V10*(0.3048/12.)**3.

END $ "OF DERIVATIVE"

DISCRETE MOUNT
PROCEDURAL

" DECOUPLER GAP DYNAMICS"

" PRES=.TRUE.; VACUUM PRESSURE STAGE"
" PRES=.FALSE.; COMPRESSIVE PRESSURE STAGE"
" CPLD=.TRUE.; THE DECOUPLER HAS BEEN COUPLED"
" CPLD=.FALSE.; THE DECOUPLER IS MOVING"

IF(CPLD) GO TO 200
  " DECOUPLER IS COUPLED"
  GAP=VGAP $ CPLD=.TRUE. $ PRES=.NOT.PRES
  LLCD3=LCODEC$ LI3=LENGTH(LLCD3,ACODEC)*1.E-3
  DI3=DI(AACODEC)*1.E-3
  A13=PHI*DI3**2./4.
  XDKEEP=XE-XS $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
  Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
GO TO 300
200.. CONTINUE
IF (DGAP.LT.0.1) GO TO 300
  " DECOUPLER IS DECOUPLED"
  CPLD=.FALSE.
  LLCD3=LCODEC$ LI3=LENGTH(LLCD3,ACODEC)*1.E-3
  DI3=DI(AACODEC)*1.E-3
  A13=PHI*DI3**2./4.
  XDKEEP=XE-XS $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
  Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
300.. CONTINUE
CALL LOGD(TRUE.)
END $ "OF PROCEDURAL"
END $ "OF DISCRETE"

IF (T.LT.TS2) GO TO 80
XDKP=XDKEEP*1000.
WRITE(2,20) T,XD,XDKP,P1,P2,CPLD,PRES
20.. FORMAT(F8.5,F8.5,F8.5,2E15.7,2I3)
80.. CONTINUE
WRITE(20,40) T
WRITE(21,40) XD
WRITE(23,40) XEDOT2
WRITE(24,40) XSDOT2
WRITE(25,40) P1
WRITE(26,40) P2
WRITE(27,40) XS
WRITE(28,40) Q1P
40. FORMAT (E15.5)

TERM(T.GE.TF)

END $ "END OF DYNAMIC"
END $ "OF PROGRAM"

SUBROUTINE ASS

OPEN (UNIT=2, FILE='PRC.LOG',STATUS='UNKNOWN')
CLOSE (UNIT=2, STATUS='DELETE')
OPEN (UNIT=2, FILE='PRC.LOG',STATUS='NEW',FORM='FORMATTED')

WRITE(2,310)
310. FORMAT(' TIME XD XDKEEP...
 P1 P2 CPLD PRES')

OPEN (UNIT=20, FILE='[.TMP]T.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=20, STATUS='DELETE')
OPEN (UNIT=20, FILE='[.TMP]T.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=21, FILE='[.TMP]XD.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=21, STATUS='DELETE')
OPEN (UNIT=21, FILE='[.TMP]XD.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=23, FILE='[.TMP]XEDOT2.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=23, STATUS='DELETE')
OPEN (UNIT=23, FILE='[.TMP]XEDOT2.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=24, FILE='[.TMP]XSDOT2.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=24, STATUS='DELETE')
OPEN (UNIT=24, FILE='[.TMP]XSDOT2.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=25, FILE='[.TMP]P1.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=25, STATUS='DELETE')
OPEN (UNIT=25, FILE='[.TMP]P1.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=26, FILE='[.TMP]P2.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=26, STATUS='DELETE')
OPEN (UNIT=26, FILE='[.TMP]P2.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=27, FILE='[.TMP]XS.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=27, STATUS='DELETE')
OPEN (UNIT=27, FILE='[.TMP]XS.DAT',STATUS='NEW',FORM='FORMATTED')

OPEN (UNIT=28, FILE='[.TMP]Q11.DAT',STATUS='UNKNOWN')
CLOSE (UNIT=28, STATUS='DELETE')
OPEN(UNIT=28,FILE=['.TMP/Q11.DAT',STATUS='NEW',FORM='FORMATTED'])

RETURN
END
PROGRAM HARMONIC RESPONSE IN FREQUENCY DOMAIN OF VEHICLE MODEL

DESCRIPTION

This program simulates the time domain response of the vehicle model installed with passive hydraulic mount. Two harmonic excitations are possible through the engine force or road profile acceleration. The initial conditions of the hydraulic mount is related with static equilibrium under the engine mass ME. Upward motion is taken positive.

If DGAP<0.1 mm, the mount is taken as the inertia track mount.
Frequency responses are calculated with reference to the last response cycles after the steady state has been reached at each excitation frequency.
As described in Section 3.5.5, the Fourier filter algorithm is implemented to extract harmonics (up to 3rd) from non sinusoidal responses.

The following frequency responses can be obtained from this program.
AXD1: phase lead of 1st harmonic XD w.r.t excitation (deg)
AXD2: phase lead of 2nd harmonic XD w.r.t excitation (deg)
AXD3: phase lead of 3rd harmonic XD w.r.t excitation (deg)
AXE1: phase lead of 1st harmonicXE w.r.t excitation (deg)
AXE2: phase lead of 2nd harmonicXE w.r.t excitation (deg)
AXE3: phase lead of 3rd harmonicXE w.r.t excitation (deg)
AXS1: phase lead of 1st harmonicXS w.r.t excitation (deg)
AXS2: phase lead of 2nd harmonicXS w.r.t excitation (deg)
AXS3: phase lead of 3rd harmonicXS w.r.t excitation (deg)
MXD1: amplitude of 1st harmonic of relative engine displacement (mm)
MXD2: amplitude of 2nd harmonic of relative engine displacement (mm)
MXD3: amplitude of 3rd harmonic of relative engine displacement (mm)
MXE1: amplitude of 1st harmonic of engine acceleration (m/s²)
MXE2: amplitude of 2nd harmonic of engine acceleration (m/s²)
MXE3: amplitude of 3rd harmonic of engine acceleration (m/s²)
MXS1: amplitude of 1st harmonic of sprung mass acceleration (m/s²)
MXS2: amplitude of 2nd harmonic of sprung mass acceleration (m/s²)
MXS3: amplitude of 3rd harmonic of sprung mass acceleration (m/s²)

LIST OF VARIABLES AND PARAMETERS

HZMN: minimum frequency in the frequency range of concern (Hz)
HZMX: maximum frequency in the frequency range of concern (Hz)
INXD(n): nth coincident component of relative engine displacement
INXE(n): nth coincident component of engine acceleration
INXS(n): nth coincident component of sprung mass acceleration
QUXD(n): nth quadrature component of relative engine displacement
QUXE(n): nth quadrature component of engine acceleration
QUXS(n): nth quadrature component of sprung mass acceleration
INITIAL
RESET('NOEVAL')
" OPEN DATA LOG FILES "
CALL ASS

" LCODE1=1 FOR CHANNEL CODE A ",
" LCODE1=14 FOR CHANNEL CODE N ",
" LCODE1=18 FOR DUAL CHANNEL DEVICE 

LOGICAL CPLD, PRES
INTEGER LCODE1, LCODED, LCODEC
CONSTANT PHI=3.141592, RHO=1059., MU=3.17E-3
CONSTANT DP=.08, GAM=1.0, LAIR=0.0005, PATM=101300.
CONSTANT AD=.0023, DGAP=0.7
CONSTANT AMP=0., FAMP=100.
CONSTANT LCODE1=1, ACODE1=2.0
CONSTANT LCODED=19, ACODED=5., LCODEC=19, ACODEC=6.
CONSTANT ME=122.3, KS=2.E4, BS=1400., MS=270.
CONSTANT HZMN=3., HZMX=20., NHAR=1.

TABLE KRTBL.1.22 ...
/1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11., ...
12., 13., 14., 15., 20., 25., 30., 35., 40., 45., 50., ...
277., 278., 280., 281., 281., 282., 283., 283., 284., 284., ...

TABLE BRTBL.1.22 ...
/1., 2., 3., 4., 5., 6., 7., 8., 9., 10., 11., ...
12., 13., 14., 15., 20., 25., 30., 35., 40., 45., 50., ...
1540., 779., 521., 396., 322., 273., 237., 211., 191., 174., 160., ...
149., 140., 132., 125., 98., 88.5, 79.9, 74., 69.3, 67., 64.5/ 

" INITIAL CONDITIONS: PIBAR(PA), V2BAR(IN**3), V1BAR(CC)"
" V1BAR IS TRANSFORMED TO IN**3 LATER"

TABLE PBAR.1.19 ...
/200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
1900., 2000., ...
101590.7, 102086.1, 102867.7, 103917.7, 105236.7, ...
106762.8, 108437.3, 110236.0, 112113.7, 114136.2, ...
116405.6, 119129.9, 122604.5, 127144.8, 133094.8, ...
140218.9, 148879.0, 158735.6, 169738.0/

TABLE V1BAR.1.19 ...
/200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
1900., 2000., ...
1.1038, 1.1704, 2.482, 3.333, 4.25, 0.5192, 0.6133, 0.7069, ...
.7986, .8922, .992, 1.1062, 1.2448, 1.4164, 1.6289, ...
1.8693, 2.1458, 2.4445, 2.7624/

TABLE V2BAR.1.19 ...
/200., 300., 400., 500., 600., 700., 800., 900., 1000., ...
1100., 1200., 1300., 1400., 1500., 1600., 1700., 1800., ...
1900., 2000., ...
.304, .454, .602, .746, .89, 1.032, 1.172, 1.312, 1.45, ...
1.588, 1.724, 1.856, 1.982, 2.1, 2.21, 2.306, 2.394, ...
2.472, 2.542/

" CHANNEL CODE= A B C D E F G H I J K L M N O P Q DUAL DECPL"
" LCODE= 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19"
" MM UNIT"

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" AREA REDUCER= FULL 1.0 2.0 2.5 3.0 4.0 5.0 6.0"

" ACODE= 0. 1.0 2.0 2.5 3.0 4.0 5.0 6.0 MM**2 UNIT"

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20., 20., 20., 20., 20., 20., 20., 20., ...
10., 10., 10., 10., 10., 10., 10., ...
10., 10., 15., 30., 40., 50., 50., ...
10., 10., 5., 30., 10., 5., 5., 7., 15., 15., ...
50., 50., 50., 50., 50., 50., 50., ...
10., 10., 10., 5., 5., 5., 5., 5., ...
5., 5., 5., 10., 10., 10., 30./

AP = PHI*DP/DP/4.
PRE = ME*9.81

DGPZ = 0.
IF (DGPZ.LT.0.1) DGPZ = 1.
IF (DGPZ.EQ.0.7) DGPZ = 2.
IF (DGPZ.EQ.1.4) DGPZ = 3.
IF (FAMP.EQ.500.) DGPZ = 4.
IF (DGPZ.EQ.0.) DGPZ = 2.

" VAIR0 = INITIAL GAS VOLUME; JUGGLE WITH LAIR TO CHANGE VAIR0"

VAIR0 = LAIR*AP

MXS = 0. $ AXS = 0. $ MXE = 0. $ AXE = 0. $ MXD = 0. $ AXD = 0.
HZ = HZMN

400.    CONTINUE
P0 = PBAR(PRE) $ V20 = V2BAR(PRE) $ V10 = V1BAR(PRE)/2.54**3.

" GEOMETRIES FOR TWO INERTIA TRACKS"
LLCD1 = LCODE1 $ L11 = LENGTH(LLCD1,ACODE1)*1.E-3 $ L12 = 0.2115-L11
D11 = DIA(ACODE1)*1.E-3 $ D12 = D11
A11 = PHI*D11**2./4. $ A12 = A11

" DECOUPLER GEOMETRY"
LLCD3 = LCODED $ L13 = LENGTH(LLCD3,ACODED)*1.E-3
D13 = DIA(ACODED)*1.E-3
A13 = PHI*D13**2./4.

CPLD = .FALSE. $ PRES = .FALSE.

KR = KRTBL(HZ)*1.E3
BR = BRKBL(HZ)
INXS = 0. $ QUXS = 0. $ INXE = 0. $ QUXE = 0. $ INXD = 0. $ QUXD = 0.

V1 = V10 $ V2 = V20
Q1 = 0. $ Q2 = 0. $ P1 = P0 $ P2 = P0
PFAC = TFPAC(HZ,DGPZ) $ NCYCLE = TCYCLE(HZ,DGPZ)
OMEGA = HZ**2.*PHI
VGAP = AD*DGAP*0.001 $ XDKEEP = 0. $ GAP = VGAP/2.
PERIOD = 1./HZ $ TF = NCYCLE*PERIOD
TS1 = TF*PERIOD $ TS2 = TF*PERIOD*2.
CINT = PERIOD/PFAC
XE = 0. $ XEDOT = 0. $ XS = 0. $ XSDOT = 0.
"FAMP=ECEN*OMEGA**2."

END

DYNAMIC

DERIVATIVE

SELp=RSW(PRES, P1-P2-100., P2-P1-100.)
SELd=RSW(CPLD, SELP, ABS(QV3)-GAP)
SCHEDULE MOUNT.XP.SELD

PROCEDURAL (P1,P2,P1E,P2E,P1EG,P2EG,PDROP1,PDROP2=V1,V2,Q1,Q2,VAIR)

P2EG=0.83420*V2**2.5-0.25117*V2**6.6+0.159415*V2**6.5
P2=P2EG*6894.41+PATM
IF(P2.LE.0.) P2=0.
P2E=P2/6894.41
IF(V1.LE.0.) GO TO 100
P1EG=-6.86729*V1+180.81844*V1**(7./6.)+19.46422*V1**2.5
P1=P1EG*6894.41+PATM
IF(P1.LE.0.) P1=0.
GO TO 110
100.. CONTINUE
P1=PATM*(VAIR0/VAIR)**GAM
110.. CONTINUE
P1E=P1/6894.41

"PRESSURE DROP ASSOCIATED WITH TURBULENT FLOW THROUGH CHANNELS"

GO TO (1,2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17,18), LCDE1
GO TO 20
1.. CONTINUE
PDROP1=0.634*1.732E12*ABS(Q1)**2.
GO TO 20
2.. CONTINUE
GO TO 20
3.. CONTINUE
GO TO 20
4.. CONTINUE
GO TO 20
5.. CONTINUE
GO TO 20
6.. CONTINUE
GO TO 20
7.. CONTINUE
GO TO 20
8.. CONTINUE
GO TO 20
9.. CONTINUE
GO TO 20
10.. CONTINUE
GO TO 20
11.. CONTINUE
GO TO 20
12.. CONTINUE
GO TO 20
13.. CONTINUE
GO TO 20
14.. CONTINUE
PDROP1=1.10*1.732E12*ABS(Q1)**2.
PDROP2=1.99*1.732E12*ABS(Q2)**2.
GO TO 20
15.. CONTINUE
PDROP1=1.10*1.732E12*ABS(Q1)**2.
PDROP2=1.99*1.732E12*ABS(Q2)**2.
GO TO 20
16.. CONTINUE
PDROP1=1.10*1.732E12*ABS(Q1)**2.
PDROP2=0.634*1.732E12*ABS(Q2)**2.
GO TO 20
17.. CONTINUE
PDROP3=(Q3/0.6/Al3)**2.*RHO/2.
END

F=FAMP*SIN(OMEGA*T)
XUDOT2=AMP*SIN(OMEGA*T)
XUDOT=-AMP/OMEGA*COS(OMEGA*T)
XU=-AMP/OMEGA**2*SIN(OMEGA*T)

XSDOT2=(-AP*(P1-P0)+KS*(XU-XS)+BS*(XUDOT-XSDOT))...
+KR*(XE-XS)+BR*(XEDOT-XSDOT))/MS
XSDOT=INTEG(XSDOT2.0.)
XS=INTEG(XSDOT,0.)

XEDOT2=(F-KR*(XE-XS)-BR*(XEDOT-XSDOT)+AP*(P1-P0))/ME
XEDOT=INTEG(XEDOT2.0.)
XE=INTEG(XEDOT,0.)

XD=(XE-XS)*1000.

" FLOW THROUGH INERTIA TRACK PATH 1"
REH1=DI1/MU*SQRT(2.*PDROP1*RHO)
I1=RHO*(L1)/AI1
Q1DOT=(P2-P1-PDROP1*SIGN(1.0, Q1))/I1
Q1=INTEG(Q1DOT,0.) $ Q1P=Q1*1.585E4
QV1=INTEG(Q1,0.)

" FLOW THROUGH INERTIA TRACK PATH 2"
REH2=DI2/MU*SQRT(2.*PDROP2*RHO)
I2=RHO*(L2)/AI2
Q2DOT=(P2-P1-PDROP2*SIGN(1.0, Q2))/I2
Q2=INTEG(Q2DOT,0.) $ Q2P=Q2*1.585E4
QV2=INTEG(Q2,0.)

" FLOW THROUGH DECOUPLER"
RE3=4.*RHO*ABS(Q3)/PHI/DI3/MU
II3=RHO*(P1/II3)/AI3
Q3DOT=(P2-P1-PDROP3*SIGN(1.0, Q3))/II3
Q3=INTEG(Q3DOT,0) $ Q3P=Q3*1.585E4
QV3=INTEG(Q3,0)

QP=Q1P+Q2P
V=QV1+QV2+QV3

V1=(-AP*(XE-XS-XDKEEP)+V)*((12./0.3048)**3+V10)
V2=V*((12./0.3048)**3+V20)
VAIR=VAIR0+AP*(XE-XS-XDKEEP)-V-V10*(0.3048/12.)**3.

" FREQUENCY RESPONSE PROCESSING"

OUTXE=RSW(T,GE,TS1, XEDOT2, 0.)
INXE1=INTEG(OUTXE*SIN(OMEGA*T),0.)
QUXE1=INTEG(OUTXE*COS(OMEGA*T),0.)
INXE2=INTEG(OUTXE*SIN(2.*OMEGA*T),0.)
QUXE2=INTEG(OUTXE*COS(2.*OMEGA*T),0.)
INXE3=INTEG(OUTXE*SIN(3.*OMEGA*T),0.)
QUXE3=INTEG(OUTXE*COS(3.*OMEGA*T),0.)

OUTXS=RSW(T,GE,TS1, XSDOT2, 0.)
INXS1=INTEG(OUTXS*SIN(OMEGA*T),0.)
QUXS1=INTEG(OUTXS*COS(OMEGA*T),0.)
INXS2=INTEG(OUTXS*SIN(2.*OMEGA*T),0.)
QUXS2=INTEG(OUTXS*COS(2.*OMEGA*T),0.)
INXS3=INTEG(OUTXS*SIN(3.*OMEGA*T),0.)
QUXS3=INTEG(OUTXS*COS(3.*OMEGA*T),0.)

OUTXD=RSW(T,GE,TS1, XD, 0.)
INXD1=INTEG(OUTXD*SIN(OMEGA*T),0.)
QUXD1=INTEG(OUTXD*COS(OMEGA*T),0.)
INXD2=INTEG(OUTXD*SIN(2.*OMEGA*T),0.)
QUXD2=INTEG(OUTXD*COS(2.*OMEGA*T),0.)
INXD3=INTEG(OUTXD*SIN(3.*OMEGA*T),0.)
QUXD3=INTEG(OUTXD*COS(3.*OMEGA*T),0.)

END "OF DERIVATIVE"

DISCRETE
MOUNT
PROCEDURAL

" DECOUPLER GAP DYNAMICS"

" PRES=.TRUE.: VACUUM PRESSURE STAGE"
" PRES=.FALSE.: COMPRESSIVE PRESSURE STAGE"
" CPLD=.TRUE.: THE DECOUPLER HAS BEEN COUPLED"
" CPLD=.FALSE.: THE DECOUPLER IS MOVING"

IF(CPLD) GO TO 200
" DECOUPLER IS COUPLED"
   GAP=VGAP $ CPLD=.TRUE. $ PRES=.NOT.PRES
   LLCD3=LCODEC5 LI3=LENGTH(LLCD3,ACODEC)*1.E-3
   DI3=DIAM(ACODEC)*1.E-3
   AI3=PHI*DI3**2./4.
   XDKEEP=XE-XS $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
GO TO 300
200.. CONTINUE
  IF (DGAP.LT.0.1) GO TO 300
  " DECOUPLER IS DECOUPLED"
  CPLD=.FALSE.
  LLCD3=LCODED $ LI3=LENGTH(LLCD3,ACODED)*1.E-3
  D13=DIA(ACODED)*1.E-3
  A13=PHI*D13**2./4.
  XDKEEP=XE-XX $ V10=V1 $ V20=V2 $ QV1=0. $ QV2=0. $ QV3=0.
  Q1=0. $ Q2=0. $ Q3=0. $ Q1P=0. $ Q2P=0. $ Q3P=0.
300.. CONTINUE
  END $ "OF PROCEDURAL"
  END $ "OF DISCRETE"

TERMT(T,GE,TF)
END $ "END OF DYNAMIC"

TERMINAL

IXE1=[NXE1/P]Q XE1=[QXE1/P]PERIOD
MXE1=SQRT(IXE1**2.+QXE1**2.)*2. $ AXE1=ATAN2(QXE1,IXE1)*180./PHI
IXE2=[NXE2/P]Q XE2=[QXE2/P]PERIOD
MXE2=SQRT(IXE2**2.+QXE2**2.)*2. $ AXE2=ATAN2(QXE2,IXE2)*180./PHI
IXE3=[NXE3/P]Q XE3=[QXE3/P]PERIOD
MXE3=SQRT(IXE3**2.+QXE3**2.)*2. $ AXE3=ATAN2(QXE3,IXE3)*180./PHI

IXS1=[NXS1/P]Q XS1=[QXS1/P]PERIOD
MXS1=SQRT(IXS1**2.+QXS1**2.)*2. $ AXS1=ATAN2(QXS1,IXS1)*180./PHI
IXS2=[NXS2/P]Q XS2=[QXS2/P]PERIOD
MXS2=SQRT(IXS2**2.+QXS2**2.)*2. $ AXS2=ATAN2(QXS2,IXS2)*180./PHI
IXS3=[NXS3/P]Q XS3=[QXS3/P]PERIOD
MXS3=SQRT(IXS3**2.+QXS3**2.)*2. $ AXS3=ATAN2(QXS3,IXS3)*180./PHI

IXD1=[NXD1/P]Q XD1=[QXD1/P]PERIOD
MXD1=SQRT(IXD1**2.+QXD1**2.)*2. $ AXD1=ATAN2(QXD1,IXD1)*180./PHI
IXD2=[NXD2/P]Q XD2=[QXD2/P]PERIOD
MXD2=SQRT(IXD2**2.+QXD2**2.)*2. $ AXD2=ATAN2(QXD2,IXD2)*180./PHI
IXD3=[NXD3/P]Q XD3=[QXD3/P]PERIOD
MXD3=SQRT(IXD3**2.+QXD3**2.)*2. $ AXD3=ATAN2(QXD3,IXD3)*180./PHI

WRITE(2,20) HZ,MXD1,AXD1,IXE1,AXE1,MXS1,AXS1
20.. FORMAT(7E11.3)
  WRITE(3,30) HZ,MXD1,AXD1,MXD2,AXD2,MXD3,AXD3
30.. FORMAT(7E11.3)
  WRITE(4,40) HZ,MXMS1,AXE1,MXMS2,AXE2,MXMS3,AXE3
40.. FORMAT(7E11.3)
  WRITE(5,50) HZ,MXS1,AXS1,MXS2,AXS2,MXS3,AXS3
50.. FORMAT(7E11.3)
  " CALL LOGD (.TRUE.)"
  DELHZ=RSW(HZ,LE.20.,1.0,2.)
  HZ=HZ+DELHZ
  IF (HZ.LE.HZMX) GO TO 400
END $ "OF TERMINAL"

END $ "OF PROGRAM"
SUBROUTINE ASS

OPEN (UNIT=2, FILE='PRC.LOG', STATUS='UNKNOWN')
CLOSE (UNIT=2, STATUS='DELETE')
OPEN (UNIT=2, FILE='PRC.LOG', STATUS='NEW', FORM='FORMATTED')

WRITE(2,310)
310. FORMAT('HZ MXD1 AXD1...
            MXE1 AXE1 MXS1 AXS1')

OPEN (UNIT=3, FILE='HXD.LOG', STATUS='UNKNOWN')
CLOSE (UNIT=3, STATUS='DELETE')
OPEN (UNIT=3, FILE='HXD.LOG', STATUS='NEW', FORM='FORMATTED')

WRITE(3,330)
330. FORMAT('HZ MXD1 AXD1...
            MXD2 AXD2 MXD3 AXD3')

OPEN (UNIT=4, FILE='HXE.LOG', STATUS='UNKNOWN')
CLOSE (UNIT=4, STATUS='DELETE')
OPEN (UNIT=4, FILE='HXE.LOG', STATUS='NEW', FORM='FORMATTED')

WRITE(4,340)
340. FORMAT('HZ MXE1 AXE1...
            MXE2 AXE2 MXE3 AXE3')

OPEN (UNIT=5, FILE='HXS.LOG', STATUS='UNKNOWN')
CLOSE (UNIT=5, STATUS='DELETE')
OPEN (UNIT=5, FILE='HXS.LOG', STATUS='NEW', FORM='FORMATTED')

WRITE(5,350)
350. FORMAT('HZ MXS1 AXS1...
            MXS2 AXS2 MXS3 AXS3')

RETURN
END
C  ***************
C  *             *
C  *  PROGRAM #5 *
C  *             *
C  ****************

C PROGRAM DPR.FOR -- DATA CONVERSION FOR TELEDYNE EXP. RESULT OF DYNAMIC
C STIFFNESS SPECTRA

C DIMENSION FREQ(100),FKDN(100),FLOS(100),CFUN(100),OMEG(100)
C DIMENSION FKEL(100),FNOM(100),TRN(100),FNFQ(100),FLOD(100)
C DIMENSION FNKD(100),ZET(100),FK(100),FK2(100)

C CHARACTER*9 DOC
C CHARACTER*80 TITLE
C TYPE *, 'ENTER FILE NAME {1-9 CHR WITHIN APOSTROPHES}'
C ACCEPT *,DO C

C OPEN (UNIT=1,FILE='[ME.GUN.HYD.DAT]'/DOC,DEFAULTFILE=':EXP',
1 STATUS='OLD')
C OPEN(UNIT=2,FILE='[ME.GUN.HYD.TMP]CTR.CPS',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE (UNIT=2,STATUS='DELETE')
C OPEN (UNIT=2,FILE='[ME.GUN.HYD.TMP]CTR.CPS',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=3,FILE='[ME.GUN.HYD.TMP]CTR.TRF',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE (UNIT=3,STATUS='DELETE')
C OPEN (UNIT=3,FILE='[ME.GUN.HYD.TMP]CTR.TRF',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=4,FILE='[ME.GUN.HYD.TMP]CTR.NFF',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE(UNIT=4,STATUS='DELETE')
C OPEN (UNIT=4,FILE='[ME.GUN.HYD.TMP]CTR.NFF',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=9,FILE='[ME.GUN.HYD.TMP]CTR.CFN',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE(UNIT=9,STATUS='DELETE')
C OPEN (UNIT=9,FILE='[ME.GUN.HYD.TMP]CTR.CFN',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=10,FILE='[ME.GUN.HYD.TMP]CTR.KEL',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE(UNIT=10,STATUS='DELETE')
C OPEN (UNIT=10,FILE='[ME.GUN.HYD.TMP]CTR.KEL',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=11,FILE='[ME.GUN.HYD.TMP]CTR.KDN',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE(UNIT=11,STATUS='DELETE')
C OPEN (UNIT=11,FILE='[ME.GUN.HYD.TMP]CTR.KDN',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=12,FILE='[ME.GUN.HYD.TMP]CTR.LOD',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C CLOSE(UNIT=12,STATUS='DELETE')
C OPEN (UNIT=12,FILE='[ME.GUN.HYD.TMP]CTR.LOD',
1 STATUS='NEW',FORM='UNFORMATTED')
C OPEN(UNIT=13,FILE='[ME.GUN.HYD.TMP]CTR.ZTA',
1 STATUS='UNKNOWN',FORM='UNFORMATTED')
C

PHI=3.141592
READ(1,'(A)') TITLE
K=INDEX(TITLE,'N')-1
READ(1,800) PREL,STSF

WRITE(8,'(A)') TITLE(1:K)
WRITE(8,820)
C23456789012345678901234567890123456789012345678901234567890123456789012

820 FORMAT(
          1'  FREQ  Kdyn  Fkdn  Zeta  CFun',
          2'  Trnf  Fnfq',/
          3'  (Hz)  (N/MM)  (Deg)  (N-S/MM)',
          4'  (Hz))
RMASS=PREL/9.806
DO 10 l=1,100,1
READ(1,800,END=20) FREQ,FKDN,FFLOD
IF (FREQ.LT.0) GO TO 20

800 FORMAT(3E10.0)
FFLOD=FFLOD*PHI/180.
OMEGG=FREQ**2.*PHI
FKDEL=FKDN*COS(FFLOS)
CFUNN=FKDN*SIN(FFLOS)/OMEGG
FFNOM=SQRT(ABS(FKDEL)**9806./PREL)
FFNFO=FFNOM/2./PHI
D1=(CFUNN*OMEGG/FFKDEL)**2
D2=(1-(RMASS*OMEGG**2)/FFKDEL/1000.)**2
TRNFF=SQRT((1.+D1)/(D2+D1))
ZETA=SQRT(1000.)*CFUNN/./SQRT(RMASS*ABS(FFKDEL))

C
FREQ(I)=FREQ
OMEG(I)=OMEGG
FKDN(I)=FKDN
FNKD(I)=FFKD/STSF
FLOS(I)=FFLOS
FLOD(I)=FFLOD
CFUN(I)=CFUNN
OMEG(I)=OMEGG
FKEL(I)=FfKEL
FNOM(I)=FFNOM
TRNF(I)=TRNF
FNFQ(I)=FFNFQ
ZETA(I)=ZETA
FK1(I)=FKD*N*COS(FFLOS)
FK2(I)=FKD*N*SIN(FFLOS)

WRITE(8,830) FREQ(I),FKDN(I),FKEL(I),ZETA(I),CFUN(I),TRNF(I),
1 FNFQ(I)
830 FORMAT(7E10.3)

10 CONTINUE
20 CONTINUE
ND=I-1

C
WRITE(7,805) ND
805 FORMAT(I3)
WRITE(7,810) (OMEG(I),FNKD(I),FLOD(I),I=1,ND)
810 FORMAT(3E15.5)
WRITE(7,1(A) 'TITLE(I=K)
WRITE(14,815) (FKDN(I),FLOD(I),FK1(I),FK2(I),FREQ(I),I=1,ND)
815 FORMAT(5F13.3)
WRITE(2) (FREQ(I),I=1,ND)
WRITE(3) (TRNF(I),I=1,ND)
WRITE(4) (FNFQ(I),I=1,ND)
WRITE(9) (CFUN(I),I=1,ND)
WRITE(10) (FKEL(I),I=1,ND)
WRITE(11) (FKDN(I),I=1,ND)
WRITE(12) (FLOD(I),I=1,ND)
WRITE(13) (ZETA(I),I=1,ND)
WRITE(15) (FK1(I),I=1,ND)
WRITE(16) (FK2(I),I=1,ND)

C
CLOSE(2)
CLOSE(3)
CLOSE(4)
CLOSE(7)
CLOSE(8)
CLOSE(9)
CLOSE(10)
CLOSE(11)
CLOSE(12)
CLOSE(13)
CLOSE(14)
CLOSE(15)
CLOSE(16)
STOP
END
PROGRAM #6

OVERLAY PLOTTING ROUTINE OF EXPERIMENT AND MATH. MODEL

PROGRAM [ME.GUN.EXP]HZ10-05.CTR
COPY [ME.GUN.ACS, TMP]T.DAT TO [ME.GUN.EXP.TMP]HZ15-10.TIME
COPY [ME.GUN.ACS, TMP]FT.DAT TO [ME.GUN.EXP.TMP]HZ15-10.FT, ETC...
HARD="TEKF", ERASE; PAGE
HZ=10; AMP=0.5;
LOAD T <[.DAT]HZ10-05.time -C1 -F -(E1PE13.4)
LOAD Y <[.DAT]HZ10-05.FT -C1 -F -(E1PE13.4)

LOAD T1 <[.TMP]HZ10-05.TIME -C1 -F -(E15.5)
LOAD Y1 <[.TMP]HZ10-05.FT -C1 -F -(E15.5)
LOAD X1 <[.TMP]HZ10-05.X -C1 -F -(E15.5)

[ROW,COL]=SIZE(T);
FOR I=1:ROW...
    X(I)=AMP*Sin(2*pi*HZ*T(I));
END;

AA=[1 1;0.5 0.6]; WINDOW(AA)
A=[0 0.15 0.05]; PLOT(A,'SCALE');
PLOT([0],[0],[0],[0],[0],[0],T,X,'NOAXES')
XLABEL('TIME (SEC)','LLLL LLL ')
YLABEL('FT (N),'L ')
A=[0 -250;0.15 250;0.05 125]; PLOT(A,'SCALE');
PLOT(T,Y,'SOLID',T1,Y1,'DASHED')
PLOT('SCALE')

AA=[1 1;0.6 0]; WINDOW(AA)
A=[-0.5 -250;0.5 250;0.5 125]; PLOT(A,'SCALE')
PLOT(X,Y,'NOAXES')
XLABEL('X (MM)', 'L LL ')
YLABEL('FT (N)', 'L ')
PLOT(X1,Y1,'DASHED')
PLOT('SCALE'); PAUSE; ERASE; PAGE

LOAD XDOT <[.TMP]HZ10-05.XDOT -C1 -F -(E15.5)
LOAD P2 <[.TMP]HZ10-05.P2 -C1 -F -(E15.5)
LOAD Q1 <[.TMP]HZ10-05.Q1 -C1 -F -(E15.5)
LOAD Q2 <[.TMP]HZ10-05.Q2 -C1 -F -(E15.5)

Q1=Q1*3.785; Q12=Q12*3.785;
AA=[1 1;0.5 0.6]; WINDOW(AA)
A=[0 0.15 0.05]; PLOT(A,'SCALE');
PLOT([0],[0],[0],[0],[0],[0],T,XDOT,'NOAXES')
XLABEL('TIME (SEC)','LLLL LLL ')
YLABEL('Q1 (LPM)')
A=[0 -8;0.15 8;0.05 4]; PLOT(A,'SCALE');
PLOT(T1,Q1,T1,Q12)
PLOT('SCALE')

P2=P2*0.001;
[ROW,COL]=SIZE(T1);
FOR i=1:ROW...
    XX1(i)=X1(i)+1.5;...
END;
AA=[1 .4;0.5 0]; WINDOW(AA)
A=[0 0.75; 0.15 2.25; 0.05 .25]; PLOT(A,'SCALE');
PLOT([0],[0],[0],[0],[0],T1,XX1,'NOAXES')
XLABEL('TIME (SEC)',LLL LLL ')
YLABEL('P2 (KPA)')
A=[0 1.12; 0.15 1.22; 0.05 5.]; PLOT(A,'SCALE');
PLOT(T1,P2)
PLOT('SCALE')
% ****************************
% *                     *
% *  PROGRAM #7  *
% *                     *
% ****************************
% Data transfer from HP 35665A to IBM-PC through SDF utilities

% TRN.M; motion transmissibility test on electrodynamic shaker
% data transfer to VAX
load dat\n13ai.dat
load dat\n13ao.dat
aai=c1; aao=c2;
load dat\n13di.dat
load dat\n13do.dat
ddi=c1; ddo=c2;
n=4;
i=1;
while n*(i-1)+1<1024
   j=n*(i-1)+1;
   t(i)=c2x(j);
   a1(i)=aai(j);
   ao(i)=aao(j);
   di(i)=ddi(j);
   do(i)=ddo(j);
   i=i+1;
end
format short e
subplot(221); plot(t,ai)
subplot(223); plot(t,ao)
subplot(222); plot(t,di,t,do)
meta plt
!gpp plt/dps/flplt2