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A New Approach to the Simulation of the Hydraulic Valve Lifter of a Car Engine

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Abstract

A 2-degree of freedom model of the hydraulic valve lifter of an internal combustion engine is investigated using different system parameters. The oil compressibility and bulk modulus are integrated into the model. System vibrations, friction forces between the parts, forces derived from the finger follower and cam system, and check valve criteria are considered within the model for the closest possible data to the real mechanism. The viscosity changes caused by the engine temperature changes, the spring constant variations resulting from the deformations and the plunger-body radial clearance changes have been investigated. Using different types of approaches, the solutions of the computer simulation are analyzed to achieve the best results.

Key words: Hydraulic lifter, Valve, Simulation, Engine, Modeling.

Introduction

The automotive industry concentrates mainly on increasing the efficiency and minimizing the maintenance requirements of internal combustion engines. Consequently, the classical valve systems used in engines are being replaced with hydraulic and electromechanical systems. Hydraulic systems increase engine performance and efficiency by improving valve timing and reducing maintenance need and improving comfort by eliminating clearances. Different types of hydraulic lifter systems can be designed. There are different designs in which both the body and the plunger are moving, along with designs in which the body is stable and the plunger is moving. In the system studied, both the body and the plunger can move freely within the block and therefore the system is of 2-DOF.

The first studies on cam lifter systems dealt with 1-DOF linear system simulation (Hornes, 1948), and then new models with higher DOF were investigated (Johnson, 1959). Later works showed that viscous damping was suitable for basic models (Hundal, 1963). In the 1980s, Pisano investigated the Coulomb friction on OHC systems (Pisano and Freudenstein, 1983; Pisano, 1984) and Hanachi analyzed the effect of vibration on engine performance (Hanachi and Freudenstein, 1986). In the late 1980s and 1990s, a cam lifter with linear valve system and nonlinear friction was investigated as a multi-DOF model (Chan and Pisano, 1987), and Pisano and Hatch added the compressibility of the oil and check valve dynamics to the model (Hatch and Pisano, 1991; Wensyang and Pisano, 1993; Hsu and Pisano, 1996; Liou *et al.*, 1998).

In this study, the hydraulic valve lifter shown in Figure 1, which can be used in different types of valve trains, has been investigated. Based on previous studies by Hatch and Pisano, more accurate results that are closer to the actual system were achieved using current technologies. The follower system studied is made up of a freely moving body placed in a block and a plunger that also moves in the body. A similar type of oil friction exists between block-body and plunger-body. Passing from the canal within the engine block, fluid flows through the orifices on the plunger and body, and fills the chambers. Oil that fills the pressure chamber in the plunger is controlled with a ball check valve located in the lower part of the plunger and the pressure change in the chamber can be adjusted. The movement of the plunger through the body is controlled with a spring connecting them. The camshaft drive directly influences the body and with the oil filling the clearance in the body and the plunger chambers a shockless movement is achieved. Through the orifice located in the upper part of the plunger, which touches the valve train, oil can reach the contact surface. Surfaces between the body-block and plunger-body are designed to produce partial contact surfaces on the top and bottom.

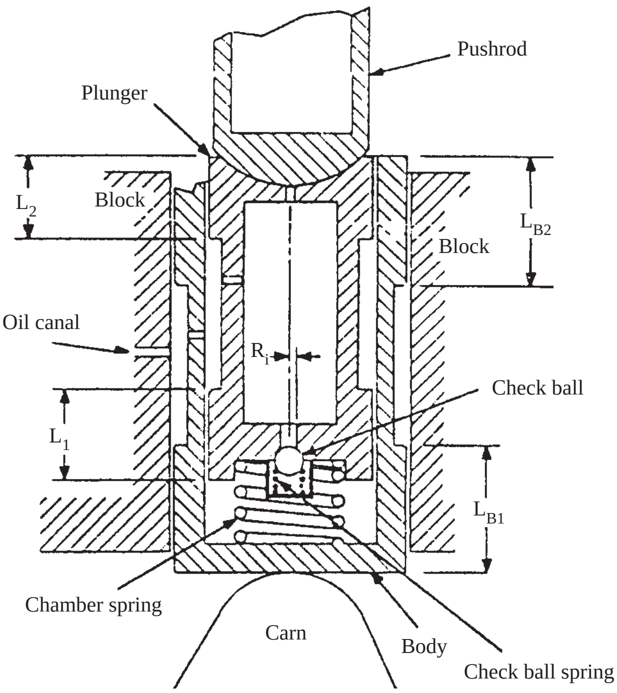

Figure 1. Hydraulic valve lifter.

This system is modeled by taking into consideration oil friction, forces affecting the system, movement of oil in the chamber, changes occurring during the opening and closing of the check valve, the amount of air entrapped by the oil, change in the bulk modulus and the cam shaft drive body (Baykara and Palabıyık, 2001a, 2001b). The spring coefficient variation caused by deformation during the life cycle of system and the plunger-body clearance change effects on the system performance have also been investigated to determine hydraulic lifter response (SAE, 1990; Baykara and Palabıyık, 2002).

Model Structure

In this analysis the oil will be treated as compressible in the oil chamber only and the compressibility will be disregarded in the modeling of the annular regions between parts. Inertial effects in the oil will not be considered, and the oil viscosity will be assumed to be constant for each simulation. The compliance of the lifter body compared to the compliance of oil in the chamber will be ignored. The dimensions of the plunger and body relative to the radial clearances shall be regarded as large enough so that gap flow can be assumed to occur between 2 parallel plates, and as a result the flow is laminar. The effects of the engine canal on oil pressure will be neglected because the pressure is less than nominal after passage through the oil canals and the orifices in the body and plunger, and has a small effect on lifter behavior.

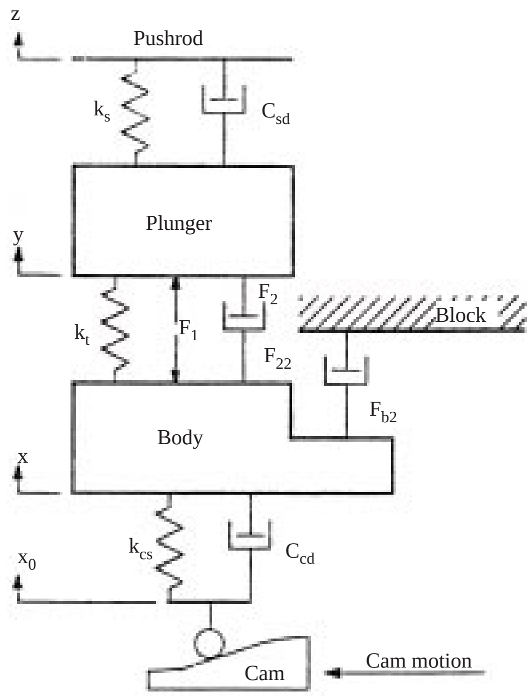

Figure 2. Two DOF system model.

The plunger and the body are assumed to be lumped masses, and to be centered in their bores. All forces are assumed to be coaxial and not to generate any moments. A damper represents the shear damping in the gap between the plunger and the body.

In Figure 2,F_1, the force on the plunger due to the oil compression for the check-valve closed and compressible oil in the chamber, can be written as

$$F_1 = A_p \Delta P = A_p \left[L_1 \left(-\frac{dP}{dx} \right) \right] \tag{1}$$

where the pressure gradient is

$$-\frac{dP}{dx} = \frac{12\mu}{h^2} \left[\frac{-A_p\left(v_p - v_c\right)}{2\pi R_p h} - \frac{v_p}{2} \right] \tag{2}$$

Substituting the value in the first equation gives

$$F_1 = -\frac{6\mu\pi R_p^2 L_1}{h^3}\left(R_p + h\right) v_p + \frac{6\mu\pi R_p^3 L_1}{h^3} v_c \tag{3}$$

where A_p is the cross-sectional area of the plunger, v_p the plunger's velocity relative to the body, h radial clearance between the plunger and the body, R_p the plunger's external radius, μ the oil's dynamic viscosity, v_c the plunger's relative velocity component due to oil compression and L_1 the lower shear contact length between the plunger and the body. The coefficients of the above equation can be defined as follows for simplicity:

$$C_{D1} = \frac{6\mu\pi R_p^2 L_1}{h^3}\left(R_p + h\right) \tag{4}$$

$$C_{Dc} = \frac{6\mu\pi R_p^3 L_1}{h^3} \tag{5}$$

Bulk modulus K can be added to Eq. 3 resulting in

$$F_1 + \frac{L_c}{K A_p} C_{Dc} \frac{dF_1}{dt} = -C_{D1} v_p \tag{6}$$

where A_p is the plunger's cross-sectional area, and L_c the chamber's height. Forces on the plunger are

$$F_1 = m_t \ddot{y} + F_{sd} + F_{ks} - F_{kt} - F_2 \tag{7}$$

where m_t is plunger mass, F_{sd} plunger and system contact damping force, F_{ks} contact spring force and F_{kt} lifter chamber preload spring force. L_1 and L_2 being the lower and upper shear contact lengths between the plunger and the body, the shear force on the plunger is

$$F_2 = -C_{D2} v_p + \frac{1}{R} F_1 \tag{8}$$

where C_{D2}=$\alpha R(1+L)$, α=$2\pi\mu L_1$, $R = R_p/h$ and $L = L_2/L_1$.

In the following equations and in Figure 2, y and x represent plunger and body displacements, $\dot{y}$ and $\dot{x}$ are the velocities, k_s the plunger and system contact spring coefficient, C_{sd} the plunger-system contact damping coefficient, F_{ks0} system spring preload, k_t the lifter chamber spring constant and L_{c0} the lifter chamber height corresponding to F_{kt0} chamber spring preload. One can write

$$v_p = \dot{y} - \dot{x}, \quad L_c = y - x$$

$$\begin{aligned} & C_0\left(y - x\right)\Big[m_t \dddot{y} + C_{D4}\ddot{y} + C_k \dot{y} - C_{D2}\ddot{x} - k_t \dot{x} \\ & -C_{sd}\ddot{z} - k_s \dot{z}\Big] + m_t \ddot{y} + C_{D5}\dot{y} + C_k y - C_D \dot{x} \\ & -k_t x - C_{sd}\dot{z} - k_s z + C_1 = 0 \end{aligned} \tag{9}$$

where $\dddot{y}$ and $\ddot{y}$ are respectively the plunger jerk and acceleration and

$$C_0 = \frac{C_{Dc}}{K A_p} \tag{10}$$

$$C_k = k_s + k_t \tag{11}$$

$$C_D = C_{D2} + \frac{R+1}{R} C_{D1} \tag{12}$$

$$C_{D4} = C_{D2} + C_{sd} \tag{13}$$

$$C_{D5} = C_D + C_{sd} \tag{14}$$

$$C_1 = F_{ks0} - F_{kt0} - k_t L_{c0} \tag{15}$$

Due to the influence of lifter chamber pressure on fluid flow in the gap, the shear forces F_2 and F_{22} are not always equal in magnitude. An additional damper models the shear force between the base and the block. The force balance of the body is

$$-F_{kt} + F_{b2} + F_{cs} + F_{cd} - F_1 + F_{22} = m_b\ddot{x} \quad (16)$$

Here F_{b2} is the viscous shear force between the body and the block, F_{cs} the cam-body contact spring force, F_{cd} the cam-body contact viscous damping force, F_{22} the shear force on the inside of the body due to fluid motion in the annular gap between the plunger and the body and m_b the mass of the body. The shear forces are

$$F_{b2} = -C_{D3}\dot{x} \quad (17)$$

$$F_{22} = C_{D2}(\dot{y} - \dot{x}) + \frac{1}{R}F_1 \quad (18)$$

where

$$C_{D3} = \alpha_b R_b(1 + L_b) \quad (19)$$

$$\alpha_b = 2\pi\mu L_{B1}; \quad R_b = R_B/h_B;$$

$$L_b = L_{B2}/L_{B1} \quad (20)$$

$L_{B1}$and L_{B2} are the lower and upper body outer lengths, R_B is the body outer radius, h_B radial gap between the body and the block. The equation of motion for the lifter body can be written as follows:

$$\begin{aligned} &C_0\,(y-x)\begin{bmatrix} m_b\dddot{x} + C_3\ddot{x} + C_{k2}\dot{x} - C_{cd}\ddot{x}_0 \\ -k_{cs}\dot{x}_0 - C_{D2}\ddot{y} - k_t\dot{y} \end{bmatrix} \\ &+m_b\ddot{x} + C_4\dot{x} + C_{k2}x - C_{cd}\dot{x}_0 - k_{cs}x_0 \\ &-C_5\dot{y} - k_t y + C_6 = 0 \end{aligned} \quad (21)$$

where x_0,$\dot{x}_0$ and $\ddot{x}_0$ are respectively the displacement, velocity and acceleration of the cam, and

$$C_3 = C_{D2} + C_{D3} + C_{cd} \quad (22)$$

$$C_{k2} = k_t + k_{cs} \quad (23)$$

$$C_4 = C_{D2} + C_{D3} + \frac{R-1}{R}C_{D1} + C_{cd} \quad (24)$$

$$C_5 = C_{D2} + \frac{R-1}{R}C_{D1} \quad (25)$$

$$C_6 = F_{kt0} + k_t L_{c0} \quad (26)$$

Oil chamber height and bulk modulus have important effects on the third order equation and one can write

$$\begin{aligned} C_0\,(y-x) &= \tfrac{C_{Dc}}{KA_p}\,(y-x) = C_{Dc}\tfrac{(y-x)}{KA_p} \\ &= C_{Dc}\left(\tfrac{1}{k_{oil}}\right) \end{aligned} \quad (27)$$

Here k_{oil} has a direct effect on the equation.

When the check-valve is open the plunger equation can be obtained as a second order linear equation as follows:

$$\begin{aligned} &C_m\ddot{y} + C_{D4}\dot{y} + C_k y - C_{D2}\dot{x} - k_t x - C_{sd}\dot{z} \\ &-k_s z + C_2 = 0 \end{aligned} \quad (28)$$

where

$$C_m = \left[m_t - \frac{R+1}{R}\left(\frac{R_p}{R_i}\right)^2 m_{ball}\right] \quad (29)$$

$$C_2 = C_1 + \frac{R+1}{R}\left(\frac{R_p}{R_i}\right)^2 F_{bs} \quad (30)$$

R_i is the radius of the plunger's internal oil passage above the check valve.

Similarly the equation of motion of the body is as follows when the check valve is open:

$$\begin{aligned} &m_b\ddot{x} + C_3\dot{x} + C_{k2}x - C_{cd}\dot{x}_0 - k_{cs}x_0 - C_7\ddot{y} \\ &-C_{D2}\dot{y} - k_t y + C_8 = 0 \end{aligned} \quad (31)$$

where

$$C_7 = \frac{1-R}{R}\left(\frac{R_p}{R_i}\right)^2 m_{ball} \tag{32}$$

$$C_8 = \frac{1-R}{R}\left(\frac{R_p}{R_i}\right)^2 F_{bs} + k_t L_{c0} + F_{kt0} \tag{33}$$

Check valve opening criterion

In the model it is assumed that the ball has the same position, velocity and acceleration as the plunger. F_{bs}, the check ball spring force, is constant and the oil in the plunger is at atmospheric pressure. Hence,

$$m_{ball}\ddot{y} = F_{bs} + F_{pc} - F_{br} \tag{34}$$

where mball is the ball mass, F_{pc} the force on the ball due to chamber pressure, F_{pr} the ball's seat reaction force. Assuming that the check valve is closed the force balance can be written as follows:

$$F_1 = \frac{R}{R+1}\left[\begin{array}{l} m_t\ddot{y} + k_x(y-z) + C_{sd}\left(\dot{y}-\dot{z}\right) \\ +F_{ks0} + k_t(y - x - L_{c0}) \\ -F_{kt0} + C_{D2}\left(\dot{y}-\dot{x}\right) \end{array}\right] \tag{35}$$

This means that under the condition $F_{br} \leq 0$ the check valve has to be open and the dynamic model has to be modified. For the model where the valve is open, the chamber pressure force on the plunger can be calculating for the condition $F_{br} = 0$.

$$F_1 = \frac{A_p}{A_B}\left[m_{ball}\ddot{y} - F_{bs}\right] \tag{36}$$

Incompressible model

Compressible models can be easily converted to the following second order linear equations by replacing the value of the bulk modulus with ∞.

$$\begin{aligned} & m_t\ddot{y} + C_{D5}\dot{y} + C_k y - C_D\dot{x} - k_t x - C_{sd}\dot{z} \\ & -k_s z + C_1 = 0 \end{aligned} \tag{37}$$

$$\begin{aligned} & m_b\ddot{x} + C_4\dot{x} + C_{k2}x - C_{cd}\dot{x}_0 - k_{cs}x_0 - C_5\dot{y} \\ & -k_t y + C_6 = 0 \end{aligned} \tag{38}$$

Reduction to 1-DOF

For the case of a firmly fixed body the equation of motion can be reduced to a single DOF and for compressible oil and the check valve closed, the following equation is obtained:

$$\begin{aligned} & C_0 y\left[m_t\dddot{y} + C_{D4}\ddot{y} + C_k\dot{y} - C_{sd}\ddot{z} - k_s\dot{z}\right] \\ & +m_t\ddot{y} + C_{D5}\dot{y} + C_k y \\ & -C_{sd}\dot{z} - k_s z + C_1 = 0 \end{aligned} \tag{39}$$

The equation becomes as follows if the check valve is open:

$$C_m\ddot{y} + C_{D4}\dot{y} + C_k y - C_{sd}\dot{z} - k_s z + C_2 = 0 \tag{40}$$

If the check valve is closed and oil is incompressible one obtains,

$$m_t\ddot{y} + C_{D5}\dot{y} + C_k y - C_{sd}\dot{z} - k_s z + C_1 = 0 \tag{41}$$

Simulation and Parameters

The differential equations obtained for the model (37-41) were solved using the Matlab software package developed by Mathworks Inc. and the graphics were obtained using Microsoft Excel. Every equation was solved by the 3 different approaches listed below to obtain better results and to allow the solutions to be checked. All 3 sets of results were compared to the experimental results of the previous study done by Hatch and Pisano and the best data were transformed into the graphs shown in the following section. This procedure was followed in order to obtain more reliable results and for a better comparison with other studies on this subject.

The approaches used in Matlab are the ODE23, LSIM and SIMULINK packages.

ODE23: Higher order differential equations were reduced and applied to the ODE23 package. The results were calculated using the Runge-Kutta method.

LSIM: The equations were converted using the Laplace transformation method and the initial conditions in state-space form were applied to solve the problem.

SIMULINK: Differential equations in Laplace form were solved with the developed block diagram by arranging the input and output values.

During the simulation process, the valve opening criterion determines which differential equation will be used and finally, depending on the position of the valve, the data collected from each equation were integrated into a single graphic for every scenario. Initial velocities and accelerations were zero in all simulations. For the leak-down test it is assumed that the body is fixed, then the model is 1-DOF and a downward constant force is applied to the top of the plunger. The simulation time is 100 ms. The pump-up simulation was held during the open and closed positions of the check valve using the 1-DOF model. The cases with compressible and incompressible oil were considered. A simulation period of 40 ms and a camshaft speed of 2500 rpm were used. The system displacement driving function was taken to be $-600e\text{-}3\sin\omega t$. The simulation parameters can be found in Tables 1-3. The numerical values of viscosity, spring load and radial clearance were changed for each simulation and compared to the previous data.

In the first mode of plunger displacement, called the leak-down period, where the check valve is closed and the system is under compression, the best data set is obtained by using the LSIM package; but in the second mode, called the pump-up period, where the check valve is open and the plunger moves upward, better results are achieved with the ODE23 package. The difference between the results obtained from the simulations with different packages depends on the algorithms used in the Matlab applications.

Table 1. Simulation parameters.

Description	Abbr.	Leak-down	Pump-Up	Transient
Plunger-system contact spring constant	k_s	0	1e6 N/m	0
Plunger-system contact spring preload force	F_{ks0}	222 N	34.7 N	222 N
Bulk modulus	K	1.38e9 Pa	1.38e9/3.72e8 Pa	1.38e9 Pa
Plunger mass	m_t	0.5 N	0.5 N	0.5 N
Body mass	m_b	0.583 N	0.583 N	0
Plunger external radius	R_p	7.25 mm	7.25 mm	7.25 mm
Plunger check ball orifice radius	R_i	1.79 mm	1.79 mm	1.79 mm
Plunger check ball mass	m_{ball}	580e-5 N	580e-5 N	580e-5 N
Plunger-body radial clearance	h	25e-3 mm	25e-3 mm	25e-3 mm
Plunger body lower shear contact length	L_1	10.2 mm	10.2 mm	10.2 mm
Plunger body upper shear contact length	L_2	4.2 mm	4.2 mm	4.2 mm
Viscosity	μ	0.297 Pa.s	0.297 Pa.s	0.297 Pa.s
Chamber spring constant	k_t	5.6e3 N/m	5.6e3 N/m	5.6e3 N/m
Body-cam contact spring constant	k_{cs}	45e6 N/m	45e6 N/m	45e6 N/m
Chamber height	L_{c0}	13.75 mm	13.75 mm	13.75 mm
Plunger-system contact damping coefficient	C_{sd}	0	0	0
Check ball spring force	F_{bs}	0.21 N	0.21 N	0.21 N
Body-cam contact damping coefficient	C_{cd}	749 N.s/m	749 N.s/m	749 N.s/m
Body-block shear contact length ratio	L_b	0.63	0.63	0.63
Block-body lower contact length	L_{b1}	16.5 mm	16.5 mm	16.5 mm
Block-body upper contact length	L_{b2}	10.5 mm	10.5 mm	10.5 mm
Body-block radial clearance	h_b	25e-3 mm	25e-3 mm	25e-3 mm
Body external radius	R_b	10.7 mm	10.7 mm	10.7 mm

Table 2. Oil viscosity values.

Engine Oil	Density	50 ° C	100 ° C
BP Super Visco-Static 20W-50	0.889kg/dm^3	75.7cSt	17.8cSt
Esso Uniflo 10W-50	0.878kg/dm^3	75.5cSt	17.2cSt

Table 3. Valve spring load change and the tolerance change in the annular gap between the body and the plunger.

Valve Spring Load	Radial Clearance
100%	25 microns
90%	30 microns

Discussion of Results

The leak-down simulation results for incompressible oil are displayed in Figure 3 in a time scale. The system displacement period is 24 ms and during each period the check valve opens once and closes once. Starting from the initial position at 100 ms, a linear displacement is observed for a duration of more than 4 periods. This means that oil passes from the orifices between the plunger and the body with constant velocity and causes a linear displacement.

The system responses for incompressible oil and for compressible oil with 0% and 1% air are depicted in Figures 4 and 5, respectively. Simulation times in these simulations were 40 ms, which is approximately 2 periods. The 3 steps of displacement can easily be observed in these figures. During the first step, which lasts about 12 ms, the system moves downward from reference zero and during this step the check valve is closed. The plunger is under the downward pressure caused by the system displacement. In Figure 5, the downward displacement depends on the compressibility of the oil, and vibrations with small amplitudes caused by the transient behavior are observed. After 12 ms, the plunger returns to the reference zero position and moves upward. After this the pressure level in the oil chamber decreases and allows the check valve to open. During the second step, the system rises to the maximum position of approximately 600 μ with an increasing velocity and finally the velocity decreases causing the check valve to close.

In Figure 4, the response of the system with the incompressible oil has no displacement during the first step, because the plunger is under the system force caused by the valve train and the oil is incompressible. After 12 ms the check valve opens and the plunger moves upward to the maximum point of the stroke and it does not move downward due to the incompressibility of the oil. Here the displacement is the absolute displacement of the plunger.

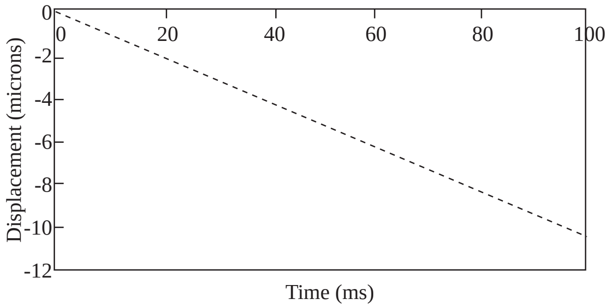

Figure 3. Effect of leak-down on plunger motion for incompressible oil.

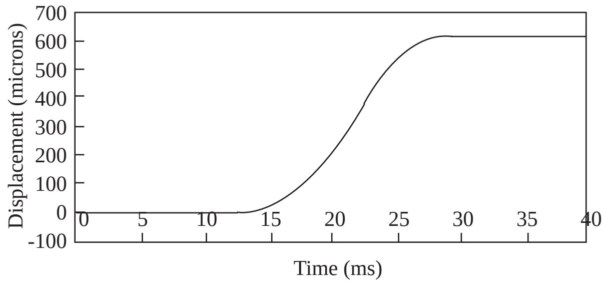

Figure 4. Plunger responses for incompressible oil.

In Figure 5, it should be noted that the velocity decreases and the system response remains stable when the check valve opens after 12 ms. This phenomenon is caused by the pressure stabilization within the mechanism after the compression effect.

In the third step the plunger moves downward under the valve spring forces, but under the effect of leak-down, a decrease in the amplitude appears. After 40 ms the system returns to the highest position. In the simulation for compressible oil, the importance of the bulk modulus can be easily seen from the differences in displacements between the diagrams for no air and 1% air. In the case of oil without air, the plunger moves downward less than 50 μ

but with the oil with 1% air the movement is more than 100 μ. The system damping changes strongly due to the bulk modulus effect, which will influence the response of the whole valve train system more strongly.

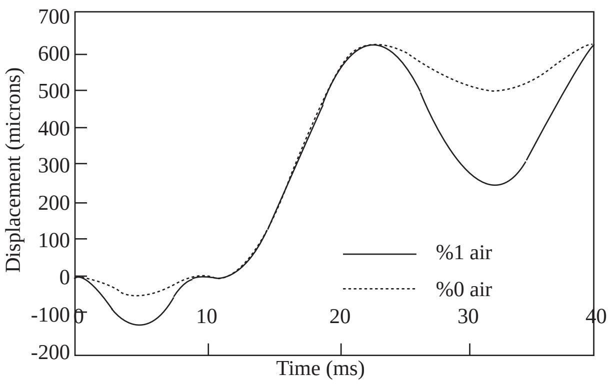

Figure 5. Plunger displacements for compressible oil.

The system studied by Pisano and Hatch has similar characteristics but the velocity change differs during the check valve opening and closing periods from the system studied here. Initially, during the first step, the system's downward motion has a more notable velocity change, because of the compressibility and the forces on the system. At the beginning the velocity is low, then it increases and at 9 ms the plunger slows down. It can be seen that the plunger remains stable for a while after 12 ms because of compensation. The velocity difference is also observed during the second step. During the second step a short period of compression can be seen and the oil flow period causes a delay on the plunger displacement different from that in the other simulation. In Figure 5, the small high frequency wiggles are less than those Pisano's study. During the third step the downward motion amplitude originating from the compressibility of oil is higher both for oil with 0% and 1% entrapped air.

In general, previous studies on hydraulic lifters were not concerned with oil viscosity changes related to engine temperature. Oil viscosity has an important effect on the vibration damping of the hydraulic lifter system of an internal combustion engine and changes directly with engine temperature.

For comparisons, first a simulation was carried out with SAE30 oil in 20 °C, similar to previous studies. The data sets obtained for the body and plunger are shown in Figure 6. Body and plunger motions have the same phase but naturally the amplitudes are different. The system response shows that there are important vibrations during the transient period despite the cam contact.

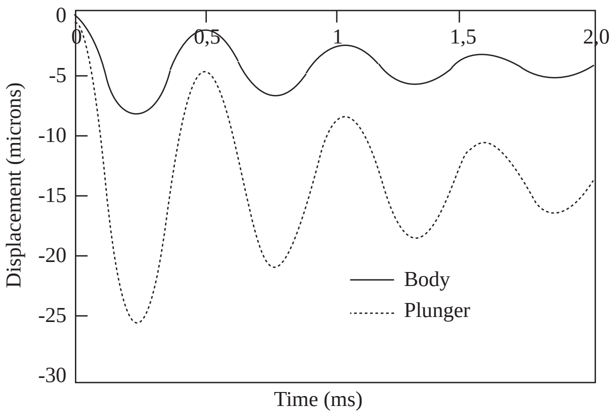

Figure 6. Plunger and body displacements within 2 ms for SAE30 oil at 20 °C.

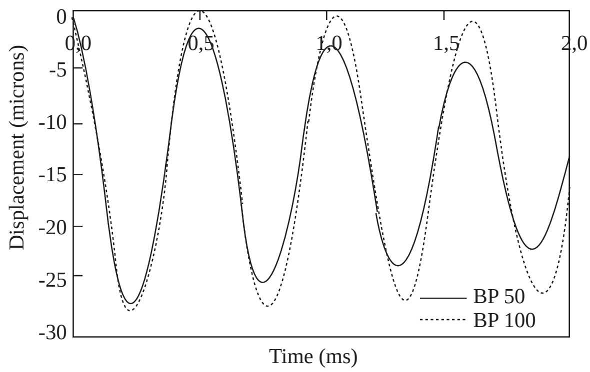

Figure 7. Plunger displacements for BP Super Visco-Static 20W oil at 50 and 100 °C.

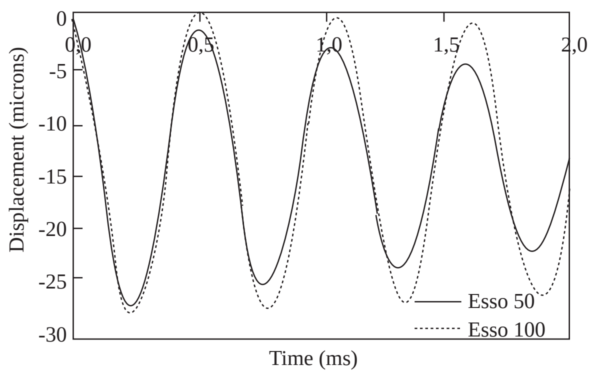

Figure 8. Plunger displacements for Esso Uniflo 10W-50 oil at 50 and 100 °C.

During the second step, the same program was run for the plunger using viscosity values of BP Super Visco-Static 20W-50 oil at 50 and 100 °C. The procedure could be repeated for the body but it is omitted because the response would have similar characteristics. It can be easily seen in Figure 7 that the viscosity change has an important effect on the system response. At higher temperatures, the viscosity value decreases and higher amplitudes are obtained. The phase is not the same at higher temperatures either. This result proves the importance of the viscosity change in a hydraulic lifter.

In order to double check the results, the same procedure was repeated with Esso Uniflo 10W-50 oil at 50 and 100 °C and the results are given in Figure 8. In this case, the viscosity values of the oils at the same temperatures are close to each other, and hence the differences between the system responses are also small.

Under high temperature and load conditions, variations in the spring coefficient and the radial clearance between the body and the plunger are general problems encountered during the life cycle of an internal combustion engine where hydraulic valve lifter systems are assumed to compensate for valve train clearances. In the last part of the study, the reaction of the hydraulic lifter to decreasing valve spring coefficient values and increasing plunger-body sliding area tolerance values is discussed. In order to see the difference clearly, the deformation and clearance ratios have been investigated for higher values of these parameters.

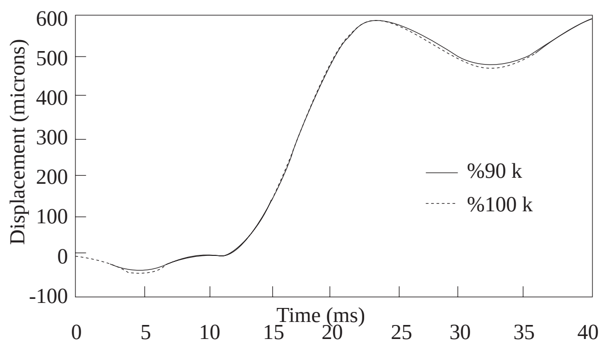

Figure 9. Spring coefficient effect on plunger response.

In Figure 9, the system responses of the lifter for 100% and 90% performances of the valve spring and for compressible oil are shown as functions of time. The simulation period is 40 ms. This diagram shows that the displacement for the 90% spring load case is less than that for the full load case during the first 12 ms period. This is expected because the load on the plunger is reduced. During the second period the displacements have approximately similar characteristics with negligible differences between them. During the third step the curve is similar in shape to that the first period, which is also expected. It can be concluded from the graph that the hydraulic lifter system is a compensator for valve train system clearances, because at the end of each period the system reaches the targeted positions without any time delay.

Comparing the 2 curves shown in Figure 10, given for 25 μ and 30 μ plunger-body radial clearances, the importance of the oil flow between the parts and the leak-down can be easily determined. During the compression period downward displacement reaches 180 μ. The bulk modulus effects for oil with 0% and 1% entrapped air, which cause displacements of 40 and 110 μ, respectively, as shown in Figure 5, indicate that this tolerance change has a notable effect on system response. During the third period the system reaches the 200 μ level, which is a significant change compared to the 350 μ obtained in Figure 5 for the bulk modulus example.

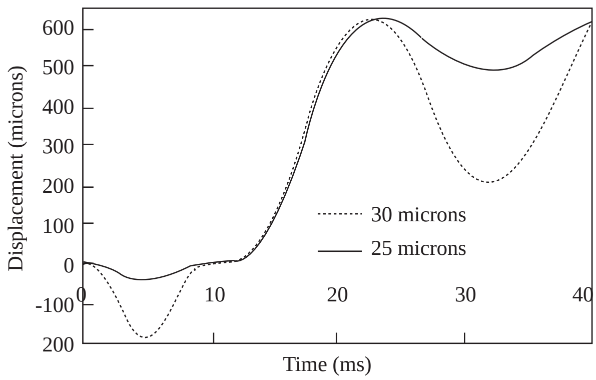

Figure 10. Effect of clearance change between the plunger and the body on plunger displacement.

Conclusion

The system investigated by Hatch and Pisano has similar characteristics but the velocity change differs during the initial motion, check valve opening and closing periods from the system studied here. Bulk modulus is shown to have an important effect on compression, which is in agreement with the work of other scientists.

Furthermore, it is demonstrated that the effect of the viscosity related to the engine temperature is important during the transient period.

Spring load has a negligible effect on plunger response but the tolerance change causes an important deviation in the negative displacement of the plunger, which is affected by the leak-down between the body and the plunger.

The simulation technique used for the model indicates better results than those obtained experimentally by Pisano and Hatch.

It can be concluded that the hydraulic lifter is a compensator for valve train system clearances, because at the end of each period the system reaches the targeted positions without any time delay. During the leak-down period, the compression level changes for every system response simulation, depending on the system parameters, but the plunger reaches the stroke limit in the appropriate time.

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Nomenclature

A_p	plunger cross-sectional area
C_{cd}	cam/body contact damping coefficient
C_{sd}	plunger/system contact damping coefficient
F_{b2}	force on body due the fluid shear stress in the annular gap between body and plunger
F_{br}	check ball reaction force
F_{bs}	check ball spring force
F_1	force due to oil compression
F_2	force due to fluid shear in annular gap between body and plunger on plunger
F_{22}	force due to fluid shear in annular gap between body and plunger on body
F_{cd}	body/cam contact damping force
F_{cs}	body/cam contact spring force
F_{ks}	plunger/system contact spring force
F_{ks0}	plunger/system contact spring preload force
F_{kt}	chamber spring force
F_{kt0}	chamber spring preload force
F_{pc}	force due to chamber pressure on check ball
F_{sd}	plunger/system contact damping force
h	plunger/body radial clearance
h_B	body/block radial clearance
K	bulk modulus
k_{cs}	body/cam contact spring constant
k_{oil}	oil spring constant
k_s	plunger/system contact spring constant
k_t	chamber spring constant
$L_{1,2}$	lower and upper plunger/body contact lengths
$L_{B1,B2}$	lower and upper block/body contact lengths
L_c	lifter chamber height
m_{ball}	check ball mass
m_b	body mass
m_t	plunger mass
R_B	body external radius
R_i	check ball orifice radius
R_p	plunger external radius
x	body displacement
x_0	cam displacement
y	plunger displacement
z	system displacement
μ	oil dynamic viscosity
$\tau_{1,2}$	fluid shear stress on lower and upper plunger/body length

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