ADAMS Model and Analysis of the GM 4L60-E Line Pressure Control

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ABSTRACT

The hydraulic systems of automatic transmissions may develop pressure fluctuations that can lead to low frequency airborne hydraulic noise. Line pressure instability at higher operating speeds has been identified as the source of this problem. In order to better study this phenomenon, a complete ADAMS model of the GM 4L60-E Variable Displacement Vane Pump (VDVP) was developed to perform dynamic analyses. The virtual prototype is capable of integrating both the mechanical and fluid components of the system simultaneously so that their interaction can be investigated. The completed model has been used extensively to study the stability of the GM vane pump. The line pressure fluctuations were analyzed by investigating the consequences of variations in different pump parameters. The effects of dimensional variation of mechanical parts, variations in fluid properties, and modification of the rotational speed of the vane were all considered.

INTRODUCTION

Variable-displacement pumps are widely used in a number of systems with varying-flow requirements. The popularity of these types of pumps is due to their lower energy consumption as compared to fixed-displacement pumps. Consequently, General Motors has been using variable-displacement vane pumps extensively in their automatic transmissions [1].

Automatic transmissions often have the problem of pressure fluctuations in the hydraulic portion of the system. These fluctuations not only overload and fatigue the system components, but they may also ultimately lead to unacceptable drivability and performance [1]. Low frequency airborne hydraulic noise ("choofing") that is detectable inside the vehicle is one of the specific problems that has been encountered.

The pressure instability dilemma can be corrected in fixed-displacement pumps by damping the regulator spool properly [1]. When using variable-displacement pumps, however, the pressure fluctuation problem cannot be corrected as easily. Because the hydraulic behavior and the dynamics of the pump are related in a variable-displacement pump, a complete dynamic analysis is necessary to study the pressure fluctuations.

The ADAMS model of the GM variable-displacement vane pump and line pressure control system incorporates both mechanical and hydraulic components to comprise the complete system. The mechanical portion includes the pressure regulator valve, booster valve, vane, and slide parts as well as a number of springs. The converter feed, decrease-chamber, and line make up the hydraulic portion of the pressure-regulation circuit. These components are modeled by utilizing the differential equations which describe the fluid dynamics of the system. Figure 1 shows a schematic of the complete ADAMS model.

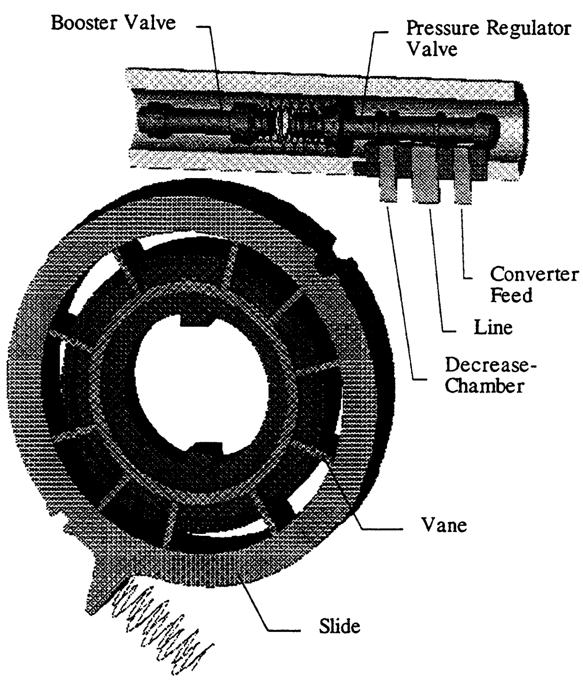

Figure 1: ADAMS Vane Pump Model

HYDRAULIC VANE PUMP

Pressure Regulator Valve

The pressure regulator valve is the pressure controlling mechanism in the variable-displacement vane pump system. Because of various disturbances and load variations, the pressure regulator valve must utilize feedback in order to sustain a fixed line pressure. The feedback is provided by the line pressure, which moderates the output flow of the pump.

When the vane pump is in use, there are two possible modes of operation of the pressure regulator valve. The flow can be directed from the line to the decrease-chamber or from the decrease-chamber to an exhaust port. The mode in which the system operates is determined by the leakage characteristics of the decrease chamber.

The motion of the pressure regulator valve is primarily dictated by pressure differences as well as spring forces. There are two springs which affect the movement of the valve. One acts between the pressure regulator valve and the outer casing, while the other acts between the pressure regulator valve and the booster valve. The equation of motion for the pressure regulator valve is given by [1]:

$$m_s\ddot{x} + b_s\dot{x} + k_s x = \Delta P_l A_s$$

where

m_s = pressure regulator valve mass

b_s = viscous damping of pressure regulator valve

k_s = total spring rate on pressure regulator valve

ΔP_l = change in line pressure

A_s = pressure regulator valve feedback area

Booster Valve

The booster valve has a significant influence on the system performance because of its affect on the movement of the pressure regulator valve. Motion of the booster valve is transferred to the pressure regulator valve through the spring that acts between them. The movement of the booster valve is largely dictated by a pressure force due to an engine torque signal. The magnitude of the torque signal, and thus the pressure force, changes with varying engine speeds. The equation of motion describing the booster valve's movement is given by:

$$m_b\ddot{x} + b_b\dot{x} + k_b x = F_b$$

where

m_b = booster valve mass

b_b = viscous damping of booster valve

k_b = spring rate on booster valve

F_b = force on booster valve due to torque signal

Slide and Vane

The slide and vane parts of the variable displacement vane pump also have a major effect on the behavior of the system. The motion of these components influences some of the pump's flow characteristics.

The major area of interest concerns the effects that the slide and the vane have on the decrease chamber flow. The amount of flow that goes into and out of this chamber is largely determined by the eccentricity of the slide. Furthermore, the flow is also affected by the engine speed and thus the RPM of the vane.

The motion of the slide can be described by a complex relation which takes into account different system pressures, the fluid characteristics, and the effects of the priming springs attached to the bottom of the slide. The effects of the vane can also be lumped into the relation, and thus the equation of motion for the slide is given by [1]:

$$J_P\ddot{\theta} + B_P\dot{\theta} + K_P R_P^2\theta = f(\theta, F_0, RPM, P_l, P_d)$$

where

J_P = moment of inertia of the slide about the pivot point

B_P = viscous damping coefficient of the slide

K_P = total priming spring rate

R_P = distance from pivot point to central axis of priming springs

θ = eccentricity of the slide

F_0 = force applied to slide by priming springs at $\theta = 0$

RPM = rotational speed of the vane

P_l = line pressure

P_d = decrease pressure

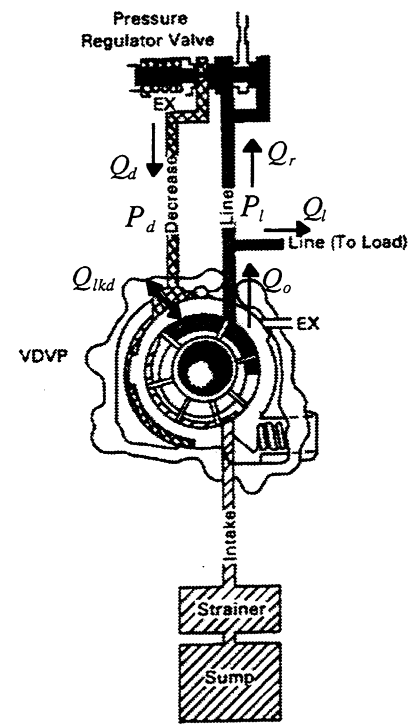

Figure 2: VDVP Fluid Model

FLUID MODEL

The variable-displacement vane pump dynamics and the system's hydraulic behavior are very closely interrelated. Therefore, in order to accurately predict the pump's performance, the model must include all of the fluid dynamic considerations.

The line and decrease pressures in the vane pump and line pressure control system have a significant effect on the pump's dynamics. The complete hydraulic description of the system can be incorporated in the differential equations which define these quantities. The differential equations describe the line and decrease pressures in terms of various system flow quantities (Figure 2). The relation for the line pressure is given by [1]:

$$\dot{P}_l = \frac{\beta}{V_l}(Q_o - Q_v - Q_l + A_v\dot{x})$$

where

P_l = line pressure

β = bulk modulus

V_l = main line volume

Q_o = pump output flow

Q_v = flow from the main line to the pressure regulator valve

Q_l = flow to load

A_v = pressure regulator valve cross - sectional area

$\dot{x}$ = velocity of the pressure regulator valve

The equation defining the decrease pressure is given by [1]:

$$\dot{P}_d = \frac{\beta}{V_d}\left(Q_d + a\dot{\theta} - Q_{lkd}\right)$$

where

P_d = decrease pressure

β = bulk modulus

V_d = decrease chamber volume

Q_d = flow from the pressure regulator valve to the decrease chamber

a = coefficient of dependence of decrease chamber volume on slide eccentricity

$\dot{\theta}$ = angular velocity of slide about the pivot point

Q_{lkd} = leakage flow from or into the decrease chamber

In addition to the line and decrease-chamber, the hydraulic portion of the system also includes the converter feed. In the ADAMS model, the assumption is made that the converter feed pressure is equal to the line pressure.

PARAMETER SWEEP AND STABILITY ANALYSIS

A sensitivity study on the variable displacement vane pump and line pressure control system was conducted with the ADAMS model in order to provide a design direction to reduce the system vibration and correct the "choofing" problem. The study involved identifying a number of critical variables and then varying these parameters above and below their baseline values. The variation in the line pressure was the quantity of interest during the parametric sweep because line pressure instability had been previously identified as the cause of the "choofing" dilemma. The line pressure instability was expressed as a percentage of the variation over the average regulated pressure. Table 1 shows the parameter sweep run matrix and the stability analysis results. The shaded entries denote the baseline values.

CONCLUSIONS

The critical system variables were identified from the results of the parametric sweep sensitivity study. The ADAMS model simulations showed that the variables which had the greatest effect on stability were the oil aeration (varied by changing the equivalent bulk modulus), the load area, and the coefficient of dependence of the pump capacity on theta (pump gain). Although their influence was not as significant, the pump speed and intermediate torque signal steps also had a substantial effect on the pump's stability.

The ADAMS test results can be used to help guide future design modifications for the variable displacement vane pump and line pressure control system. To reduce vibration and correct the "choofing" problem, a number of different approaches could be taken:

- Reduce the oil aeration
- Increase the load leakage
- Decrease the pump gain

Reducing the oil aeration not only caused the most significant stability improvements, but it is also the modification that can be most feasibly implemented. The equivalent bulk modulus used in the analyses takes into account both the fluid bulk modulus (which is constant) and the amount of air in the oil. This value can vary greatly depending on the amount of air that is present. Changes can easily be made in the transmission system which can help to limit the air entering the oil, and thus improve stability.

The other two correction approaches are not as desirable as reducing the oil aeration. Although increasing the load leakage and decreasing the pump gain helped to reduce vibration and correct the "choofing" problem, there are drawbacks to implementing these ideas. Decreasing the pump gain may cause the variable displacement vane pump to not meet certain system requirements, and increasing the load leakage can decrease the system's efficiency.

REFERENCES

1. Karmel, A., "Stability and Regulation of a Variable Displacement Vane-Pump". General Motors Research Laboratories, Power Systems Research Department, October 1983.

Test	A	B	C
1. Oil Aeration (Bulk Modulus)	344.75 MPa **unstable**	689.5 MPa **147%**	1379 MPa **9%**
2. Pump Speed	2000 RPM **147%**	4000 RPM **180%**	5000 RPM **182%**
3. Valve Damping	0.1 N*s/mm **unstable**	0.2 N*s/mm **147%**	0.4 N*s/mm **154%**
4. Slide Damping	750 N*s/(mm*rad) **unstable**	751 N*s/(mm*rad) **147%**	752 N*s/(mm*rad) **143%**
5. Regulator Spring Rate	0.572 N/mm **135%**	1.144 N/mm **147%**	11.44 N/mm **unstable**
6. Slide Friction	2500 N*mm **150%**	5000 N*mm **147%**	10000 N*mm **141%**
7. Decrease Leakage (equiv. orifice dia.)	0.0 mm **147%**	0.5 mm **148%**	1.0 mm **149%**
8. Intermediate Steps (torque signal)	30 psi **90%**	60 psi **127%**	90 psi **147%**
9. Load Area	1.0*Area **147%**	1.5*Area **47%**	
10. Pump Capacity Dependence on Slide Eccentricity, a	0.75*a **51%**	1.0*a **147%**	

Table 1: Stability Analysis Run Matrix