

Prediction of Vehicle Interior Noise from a Power Steering Pump using Component CAE and Measured Noise Transfer Functions of the Vehicle

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ABSTRACT

In response to the growing demand for fuel economy, we are developing a high-efficient variable displacement pump for hydraulic power steering systems. In order to develop a quiet variable displacement pump which generates lower noise for better vehicle interior sound quality, we have been developing a simulation tool which includes hydraulic analysis, vibration analysis, and vehicle interior noise analysis which combines simulation outputs and measured noise transfer functions of the targeted vehicle. This paper provides both validation results of the simulation tool and application examples to design improvement to conclude the effectiveness of the simulation tool developed.

INTRODUCTION

In response to the growing demand for fuel economy, we are developing a high-efficient variable displacement pump for hydraulic power steering systems. The variable displacement pump is designed to maintain a constant flow rate regardless of the engine-driven pump revolution speed, by mechanically controlling its volumetric displacement. This feature is beneficial for reducing the pump torque, since a conventional fixed displacement pump, which generates flow volume in proportion to engine speed, generates unnecessarily high flow volume as engine speed goes up, and demands more torque.

Figure 1 shows the simplified structure of the developed variable displacement pump. Eleven vanes (which are forming 11 vane rooms) are pushed to the cam by back pressure, and the rotor (which is holding the vanes) revolves in a counterclockwise direction in proportion to engine speed

(driven by the engine). As shown in the figure, each of the upper-half vane rooms gradually expands its volume as the rotor revolves, and hydraulic oil is suctioned from the suction port on a side plate. On the other hand, the lower-half vane rooms gradually reduce their volume, thereby pushing hydraulic oil out of the discharge port on the side plate, and the discharged oil goes through a high pressure pipe toward the power cylinder. The discharged pressure goes up in response to the driver's steering action, which squeezes the rotary valve of the steering system, supplying enough pressure (up to 10 to 15 MPa) into the power cylinder to assist steering.

The mechanism of mechanically controlling the flow rate to a constant value is as follows. At low revolution speed the cam is pushed to the extreme left by spring force and internal pressure of vane rooms, and the volumetric displacement per revolution is at the maximum, at which position the discharged flow rate increases in proportion to the pump revolution speed. As the revolution speed rises and the flow rate is over a certain level, the pressure drop at the flow control orifice generates enough pressure difference between left and right side of the control valve to move it rightward against the spring force. This opens the control valve, making high pressure oil to flow into the cam left room, and the cam moves rightward, decreasing the volumetric displacement per revolution, ending up to maintain the flow rate constant. Because the necessary flow rate for the steering assist stays almost the same when the pump revolution goes up, the variable displacement pump can reduce the unnecessary energy consumption at high revolution.

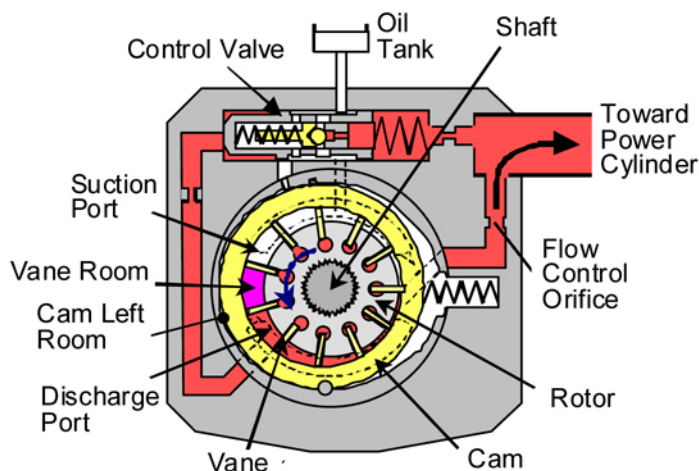


Figure 1. Structure of the variable displacement pump

Although the variable displacement pump is effective for energy saving, it has the potential to generate bigger pressure pulsation and vibration compared to a conventional fixed displacement pump. A fixed displacement pump has two sets of discharge and suction section in symmetric position about the shaft, cancelling out the shaft excitation force which derives from the pressure change at the transition section from suction to discharge, whereas the variable displacement pump cannot help employing an asymmetric port configuration fundamentally.

Also, the fixed displacement pump can reduce pressure pulsation by optimizing cam's inner profile and by adjusting the transition timing of the discharge and suction section, whereas the variable displacement pump basically should have a circle cam and the transition timing changes as the cam moves, so that it is difficult to decrease pressure pulsation in all revolution range.

With above reasons, in order to develop a quiet variable displacement pump for better vehicle interior NVH quality, it is important to predict the interior noise induced by the pressure pulsation and vibration and to decrease both in a highly balanced way by tuning several design parameters.

Power steering noise is considered to be perceived in the vehicle interior cabin through 3 different transfer paths; fluid borne path, structure borne path, and air borne path [1]. Figure 2 shows the noise transfer paths of the power steering pump. Fluid borne noise originates from pressure pulsation discharged from the power steering pump, propagating through the high pressure pipe. Structure borne noise originates from the vibration of the power steering pump, propagating through the engine mount toward chassis frame, since the power steering pump is typically mounted on the engine. Air borne noise originates from the sound radiation of the pump surface induced by the vibration, propagating through the air into the vehicle interior.

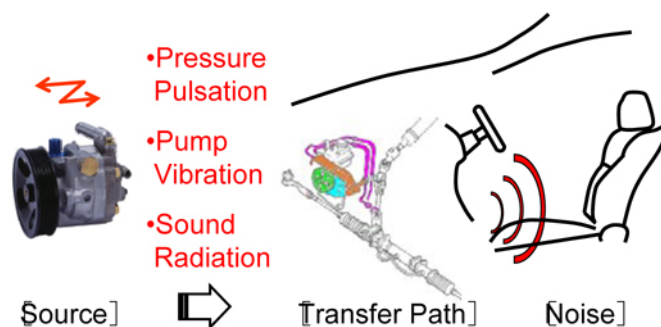


Figure 2. Noise transfer paths of a power steering pump

Noise from a power steering pump is often described as “moan” and “whine” [2]. “Moan” noise is typically related to the fluid borne noise, which has relatively low-frequency content (e.g. 100Hz - 300Hz), and sometimes becomes audible at high discharged pressure conditions (e.g. above 6 MPa). “Whine” noise is typically related to the structure borne and air borne noise, which has relatively high-frequency content (e.g. above 500Hz up to 2kHz), and sometimes becomes audible at high revolution of the pump (e.g. 1500 r/min - 3000 r/min).

Previous works [3, 4, 5] were mostly dedicated to reducing the pressure pulsation and flow ripple of the power steering pump by hydraulic analysis. These researches were much detailed and effective for fluid borne noise reduction, but it is also necessary to analyze the vibration of the pump in order to reduce the structure borne and airborne noise. One of a few literatures which deal with the vibration analysis of the pump is [6], in which a finite element model of the pump was created, and measured internal pressure load was applied to the inner surface of the pump.

Although hydraulic analysis and vibration analysis is useful itself to predict and reduce the pressure pulsation and vibration of the pump respectively, actual vehicle interior noise is affected by the characteristics of the vehicle regarding each transfer path, so we can not fully assess the NVH quality of the pump only by hydraulic or vibration analysis. In order to assess the vehicle interior noise derived from the pump beforehand and to decrease the noise by design modification while balancing the fluid borne, structure borne, and air borne noise, prediction method of vehicle interior noise is necessary.

For this purpose we have been developing a simulation technology that can predict the vehicle interior noise from the pump CAE model by combining measured transfer functions of the vehicle. The CAE model consists of the hydraulic analysis and vibration analysis of the developed variable displacement pump. As the vehicle model is not usually available from automakers and even if available it is difficult

to model the whole vehicle precisely, we will use the measured noise transfer functions of the vehicle.

As far as we know, this kind of vehicle interior noise prediction which combines power steering pump CAE model and vehicle transfer functions has not been reported yet.

OVERVIEW OF THE SIMULATION TECHNOLOGY BEING DEVELOPED

Figure 3 shows the schematic diagram of the simulation flow. Each output from the component CAE will be multiplied by a noise transfer function of the vehicle and summed up to total vehicle interior noise. Here, vehicle P/P means a transfer function of sound pressure at driver's ear position over pressure pulsation, vehicle P/F means sound pressure over transmitted force by pump vibration, and vehicle P/W means sound pressure over sound power emitted from the pump surface. These transfer functions will be measured from the targeted vehicle.

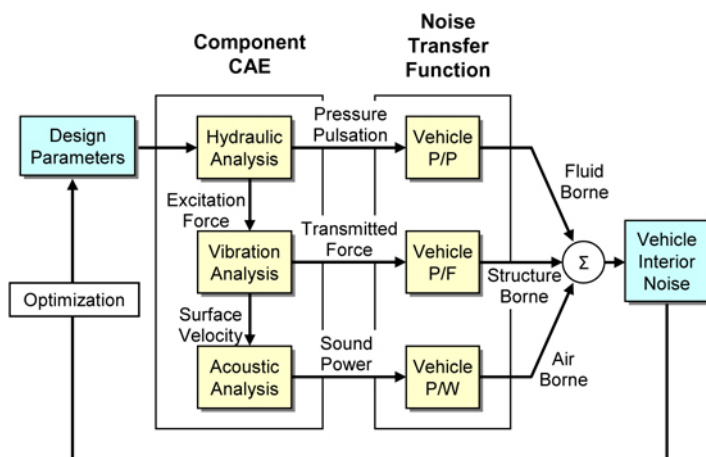


Figure 3. Schematic diagram of the simulation flow

A similar work was conducted by [7] using measured component data from the test stand and measured vehicle transfer functions, but in our work we model the component using CAE, so once we can obtain the vehicle transfer functions we can run the simulation for optimization of the design parameters to highly achieve both pump characteristics and vehicle NVH.

Only fluid borne noise and structure borne noise will be considered in this paper because from our experience these are more important and dominant paths for the variable displacement pump so that higher priority was put on these, but virtually the same approach can be taken on air borne noise of course.

HYDRAULIC ANALYSIS OF A VARIABLE DISPLACEMENT PUMP

For simulation of hydraulic systems, lumped parameter system models are often used [8, 9]. Previously, we have developed a hydraulic simulator for a high-pressure fuel pump for direct injection gasoline engine [10]. The same modeling approach has been used here.

As each of 11 vane rooms is small enough to be assumed to have uniform pressure distribution, they are modeled as a lumped volume room whose size is variable, and formulated as

$$q_1 - q_2 - \frac{dV}{dt} = \frac{V}{B} \frac{dp}{dt} \quad (1)$$

where V is the volume of the vane room, P is the pressure of the vane room, B is the bulk modulus of the hydraulic oil, q_1 is the incoming volume of the hydraulic oil into the vane room, and q_2 is the outgoing volume of the hydraulic oil out of the vane room.

Internal oil passages inside the pump can be assumed to be short pipes, so the lumped-model motion equation of the hydraulic oil flowing inside these short pipes, as well as the effect of compressibility of the hydraulic oil is respectively given by

$$\rho L \frac{dq_1}{dt} = a(p_1 - p_2) - \rho L R_f q_1 \quad (2)$$

$$q_1 - q_2 = \frac{aL}{B} \frac{dp_2}{dt} \quad (3)$$

where P_1 and P_2 is the pressure at the beginning and ending edge of the pipe respectively, q_1 and q_2 is the incoming and outgoing flow volume at each edge respectively, ρ is the density of the hydraulic oil, L is the length of the pipe, a is the cross-sectional area of the pipe, and R_f is the coefficient of flow resistance of the pipe.

Flow at orifices between oil passages inside the pump or leakage from the clearance gap between each part can be similarly given by the formula of hydraulic circuit. Motion of the cam moved by the moment balance of inner and outer surface pressure of the cam and spring force, or the motion of the control valve by the spring force and pressure difference