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# REDUCTION OF PRESSURE RIPPLES USING A PARALLEL LINE IN HYDRAULIC PIPELINE

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**ABSTRACT**–Pressure ripples, which are inevitably generated by a fluctuation of flow rate caused by a pump mechanism, include noises and vibrations in hydraulic pipeline. These noises and vibration deteriorate the stability and accuracy of hydraulic systems. The accumulator and hydraulic attenuator are normally used to reduce the pressure ripples. In this study, a parallel line is introduced to the hydraulic pipeline for the hydraulic system with a bent-axis piston pump as a method to reduce the pressure ripples. The dynamic characteristics of the hydraulic pipeline with a parallel line are analyzed by a transfer matrix in the frequency domain. The usefulness of the hydraulic pipeline with a parallel line was ascertained by experiment and simulation. The results from the experiment and simulation show that the hydraulic pipeline with a parallel line were effective in reducing the pressure ripples.

**KEY WORDS :** Pressure ripples, Flow rate, Hydraulic pipeline, Parallel line, Bent-axis piston pump, Transfer matrix

## 1. INTRODUCTION

Hydraulic systems have been widely used in industry for power supply. Hydraulic pump, which is one of the most important equipment in a hydraulic system, supplies unsteady-state flow rates to the system. This unsteady condition is due to the mechanism itself –(Yoo *et al.*, 2002). The fluctuation of the flow rates causes pressure ripples in the hydraulic pipeline. The pressure ripples includes noises and vibrations in hydraulic system components, such as connecting pipes and control valves, and they deteriorate the stability and accuracy of the system. Therefore, the equipments, such as an accumulator, hydraulic attenuator, and pressure control valve, are used to reduce the pressure caused by these ripples (Kim *et al.*, 1997). In this paper, a parallel line is introduced to the hydraulic pipeline in the hydraulic system with a bent-axis piston pump, widely used in hydraulic systems because of high pressure level, best efficiency, and low operating cost as a method to reduce pressure ripples (Jaroslav *et al.*, 1985). The dynamic characteristics of the hydraulic pipeline with a parallel line are analyzed by a transfer matrix in the frequency domain. An experiment and simulation were carried out to ascertain the ability of the hydraulic pipeline with a parallel line to reduce pressure ripples.

## 2. THEOREITICAL ANALYSIS

### 2.1. Bent-axis Piston Pump

The scheme of a bent-axis piston pump is shown in Figure 1. The cylinder bores are arranged on a circle of radius in the cylinder block, which is arranged at a particular angle to the driving shaft. The pistons execute a linear movement in the cylinder bores and are directly

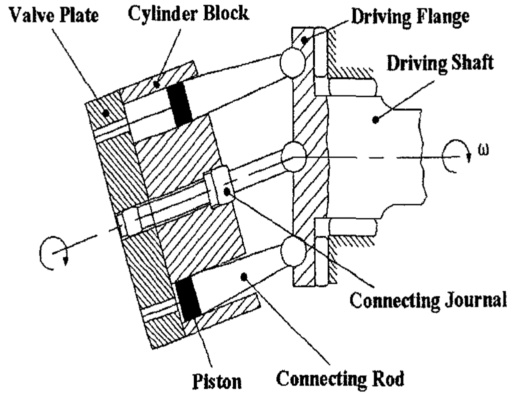

Figure 1. Configuration of a bent-axis piston pump.

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*Corresponding author. e-mail: kimkh@khu.ac.kr connected to the driving flange by ball joints. The connecting rod is used as a connecting element between the piston and driving flange. Since the cylinder block and driving flange rotate together, the cylinder block rotation and driving flange rotation must be synchronized. The displacement chambers connect with the suction and discharge port when the rotating cylinder block with its openings in the cylinder bottom slides over the fixed valve plate. While the positive displacement chamber increases (intake stroke), it connects with the suction port and the working oil is sucked into the displacement chamber. The positive displacement chamber reaches its maximum volume, and then while it decreases (discharge stroke), the pressure in the positive displacement chamber rises and the working oil flows to the pump outlet and the pump performs the displacement work.

Through the above continuous displacement work, flow rates are generated. The instantaneous flow rates with an odd number of pistons are given as follows in the interval $0 < \varphi \leq \pi/z$,

$$Q_i = \omega A R_2 \sin \sum_{i=1}^{z/2+0.5} \sin\left[\varphi + (i-1)\frac{2\pi}{z}\right] \tag{1}$$

and in the interval $\pi/z < \varphi \leq 2\pi/z$, the instantaneous flow rates are given as follows.

$$Q_i = \omega A R_2 \sin\beta \sum_{i=1}^{z/2-0.5} \sin\left[\varphi + (i-1)\frac{2\pi}{z}\right] \tag{2}$$

In case of an even number of pistons, the instantaneous flow rates are given as follows.

$$Q_i = \omega A R_2 \sin\beta \sum_{i=1}^{z/2} \sin\left[\varphi + (i-1)\frac{2\pi}{z}\right] \tag{3}$$

where $\omega$ is the angular velocity of the cylinder block, A is the area of valve plate opening to the discharge port, $R_2$ is the radius from the center of the driving flange to the ball joint, $\beta$ is the angle of inclination of the cylinder block, z is the number of pistons, and $\varphi$ is the angular position of piston.

Bent-axis piston pump has different gaps between the parts moving relative to each other. The working oil flows through these gaps due to the pressure difference. The flows also depend on kinematic viscosity and rotational speed. Figure 2. shows the leakages in a piston of a bent-axis piston pump (Jaroslav *et al.*, 1985). The flow rates $Q_r$ represent the flow rates in and out of the cylinder through the valve plate opening. The flow rate $Q_{pc}$, $Q_{cv}$ and $Q_{bs}$ denote the leakages. $Q_{pc}$ is the leakage through the gap between the piston and cylinder block;, $Q_{cv}$ is the leakage through the gap between the cylinder block and the valve plate and $Q_{bs}$ is the leakage through the bore in the piston to the ball joint. These leakages can be theoretically obtained from the solution to the Reynolds equation. The pressure gradient with respect to time is given as follows (Jaroslav *et al.*, 1985).

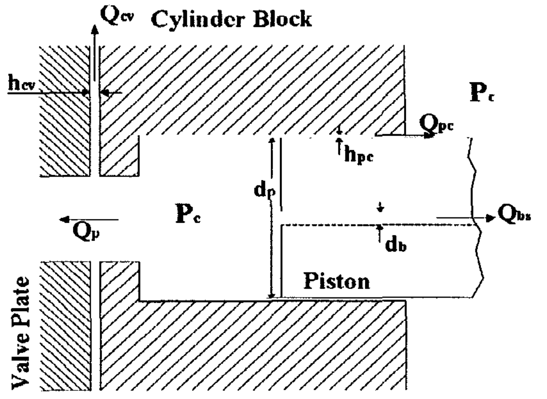

Figure 2. Leakages in piston of a bent-axis piston pump.

$$\frac{dP}{dt} = \frac{K}{V}\left(\Sigma Q - \frac{dV}{dt}\right) \tag{4}$$

To determine the discharge pressure at the pump outlet, Equation (4) is extended to Figure 3, which shows the discharge port connected to the hydraulic pipeline. Then, the equation for the discharge pressure at the pump outlet of bent axis is obtained as follows.

$$\frac{dP}{dt} = \frac{K}{V}(-Q_{in} + Q_{out}) \tag{5}$$

where, $V_h$ is the total volume of the discharge port and hydraulic pipeline. So, the length of the hydraulic

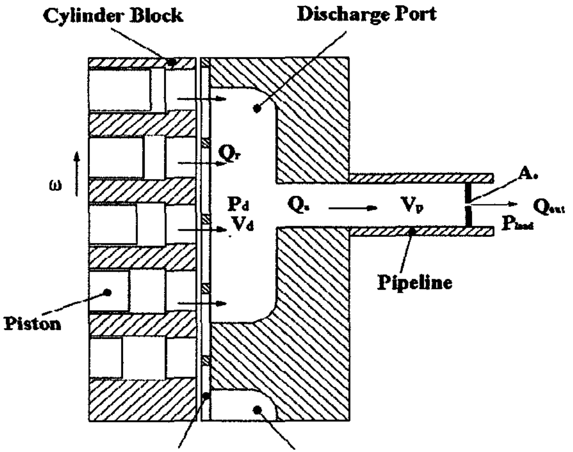

Figure 3. Scheme of discharge port for analysis of discharge pressure on the outlet of a bent-axis piston pump.

pipeline influences the discharge pressure ripples at the pump outlet. $Q_{in}$ denotes the total flow rates into the discharge port and $Q_{out}$ denotes the flow rates through the orifice valve. These flow rates are respectively given as follows.

$$Q_{in} = \sum_i Q_r \tag{6}$$

$$Q_{in} = \alpha A_h \sqrt{\frac{2}{\rho}(p - p_{load})}\,\mathrm{sgn}(p - p_{load}) \tag{7}$$

2.2. Hydrauric Pipeline with a Parallel Line

The relationships among the pressure $P_a$ and the flow rates $Q_a$ at the upstream end and the pressure $P_b$ and the flow rates $Q_b$ at the downstream end of a simple hydraulic pipeline are represented by a transfer matrix in the frequency domain (Benjamin, 1983).

$$\begin{bmatrix} P_a \\ Q_a \end{bmatrix} = \begin{bmatrix} \cosh(\gamma l) & Z_c \sinh(\gamma l) \\ 1/Z_c \sinh(\gamma l) & \cosh(\gamma l) \end{bmatrix} \begin{bmatrix} P_b \\ Q_b \end{bmatrix} \tag{8}$$

where $l$ is the length of the line, $\gamma$ and $Z_c$ are the propagation constant and characteristic impedance respectively, which are derived as follows.

$$\gamma = \frac{s}{a}\sqrt{1 + \frac{8\pi\nu}{A_p s}} \quad \cdot \tag{9}$$

$$Z_c = \frac{\rho a}{A}\sqrt{1 + \frac{8\pi\nu}{A_p s}} \tag{10}$$

where $a$ is the velocity of the pressure wave, $A_p$ is the sectional area of the line, $\rho$ is the density of oil, and $s$ is the Laplace operator.

Figure 4 shows the hydraulic pipeline with a parallel line. From continuity relations at the upstream and the down-stream junction, and assumption that the number 1 and 2 line between the junctions are simple hydraulic pipelines, the equations are obtained as follows (Yang *et al.*, 1986).

$$Q_b = Q_c + Q_d \tag{11}$$

$$Q_g = Q_e + Q_f \tag{12}$$

$$\begin{bmatrix} P_c \\ Q_c \end{bmatrix} = \begin{bmatrix} \cosh(\gamma l_2) & Z_{c2} \sinh(\gamma l_2) \\ 1/Z_{c2} \sinh(\gamma l_2) & \cosh(\gamma l_2) \end{bmatrix} \begin{bmatrix} P_e \\ Q_e \end{bmatrix} \tag{13}$$

$$\begin{bmatrix} P_d \\ Q_d \end{bmatrix} = \begin{bmatrix} \cosh(\gamma l_3) & Z_{c3} \sinh(\gamma l_3) \\ 1/Z_{c3} \sinh(\gamma l_3) & \cosh(\gamma l_3) \end{bmatrix} \begin{bmatrix} Q_f \\ Q_f \end{bmatrix} \tag{14}$$

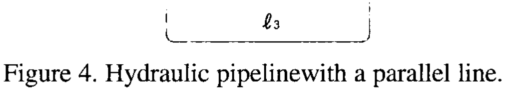

Figure 4. Hydraulic pipelinewith a parallel line.

$$\begin{vmatrix} P_b(s) \\ Q_b(s) \end{vmatrix} = \begin{vmatrix} \dfrac{Z_3S_3C_2 + Z_2S_2C_3}{Z_2S_2 + Z_3S_3} & \dfrac{Z_2Z_3S_2S_3}{Z_2S_2 + Z_3S_3} \\ \dfrac{2Z_2Z_3(C_2C_3 - 1) + (Z_2^2 + Z_3^2)S_2S_3}{Z_2Z_3(Z_2S_2 + Z_3S_3)} & \dfrac{Z_3S_3C_2 + Z_2S_2C_3}{Z_2S_2 + Z_3S_3} \end{vmatrix} \begin{vmatrix} P_g(s) \\ Q_g(s) \end{vmatrix} \tag{15}$$

Applying $P_b = P_c = P_d$ and $P_e = P_f = P_g$ at the junctions and eliminating $Q_c$, $Q_d$, $Q_e$ and $Q_f$ in equation (11), (12), (13) and (14), we can obtain the following transfer matrix which represent the dynamic characteristics of the parallel line part (Yang *et al.*, 1986).

where sinh $(\gamma l_i)$, $\cosh(\gamma l_i)$ and $Z_{ci}$ are expressed as $S_i$, $C_i$ and $Z_i$ to simplify the notation, and the subscript $i$ indicates the line number. The lines before and after the parallel line part are simple hydraulic transmission lines. The transfer matrices of the lines are obtained from the Equation (3) as follows.

$$\begin{bmatrix} P_a \\ Q_a \end{bmatrix} = \begin{bmatrix} \cosh(\gamma l_1) & Z_{c1} \sinh(\gamma l_1) \\ 1/Z_{c1} \sinh(\gamma l_1) & \cosh(\gamma l_1) \end{bmatrix} \begin{bmatrix} P_b \\ P_b \end{bmatrix} \tag{16}$$

$$\begin{bmatrix} P_g \\ Q_g \end{bmatrix} = \begin{bmatrix} \cosh(\gamma l_4) & Z_{c4} \sinh(\gamma l_4) \\ 1/Z_{c4} \sinh(\gamma l_4) & \cosh(\gamma l_4) \end{bmatrix} \begin{bmatrix} P_h \\ Q_h \end{bmatrix} \tag{17}$$

$$\begin{vmatrix} P_a(s) \\ Q_a(s) \end{vmatrix} = \begin{vmatrix} \cosh(\Gamma l_1) & Z_{c1} \sinh(\Gamma l_1) \\ 1/Z_{c1} \sinh(\Gamma l_1) & \cosh(\Gamma l_1) \end{vmatrix} \times$$

$$\begin{vmatrix} \dfrac{Z_3S_3C_2 + Z_2S_2C_3}{Z_2S_2 + Z_3S_3} & \dfrac{Z_2Z_3S_2S_3}{Z_2S_2 + Z_3S_3} \\ \dfrac{2Z_2Z_3(C_2C_3 - 1) + (Z_2^2 + Z_3^2)S_2S_3}{Z_2Z_3(Z_2S_2 + Z_3S_3)} & \dfrac{Z_3S_3C_2 + Z_2S_2C_3}{Z_2S_2 + Z_3S_3} \end{vmatrix} \times$$

$$\begin{vmatrix} \cosh(\Gamma l_4) & Z_{c4} \sinh(\Gamma l_4) \\ 1/Z_{c4} \sinh(\Gamma l_4) & \cosh(\Gamma l_4) \end{vmatrix} \begin{vmatrix} P_h(s) \\ Q_h(s) \end{vmatrix} \tag{18}$$

Substituting the Equation (15) and Equation (17) into the Equation (16) one by one, the transfer matrix, which represents the relationships between the inlet and outlet in hydraulic pipeline with a parallel line, is obtained as follows.

Pressure waves are divided in two ways at the upstream junction and arrive at the downstream junction out of phase with each other in the hydraulic pipeline with a parallel line. When the length difference between the number 1 line and number 2 line is an odd multiple of the half wavelength, the phase difference of the two pressure waves is 180°, when the waves arrive at the downstream junction (Stan Skaistis, 1988). Then the pressure ripples cancel each other completely. The equation for this behavior is as follows.

$$l_3 - l_2 = k\frac{f}{2} = k\frac{a}{2f}, \quad k = 1, 3, 5\cdots \tag{19}$$

where $\lambda$ and $f$ are the wavelength and frequency of the

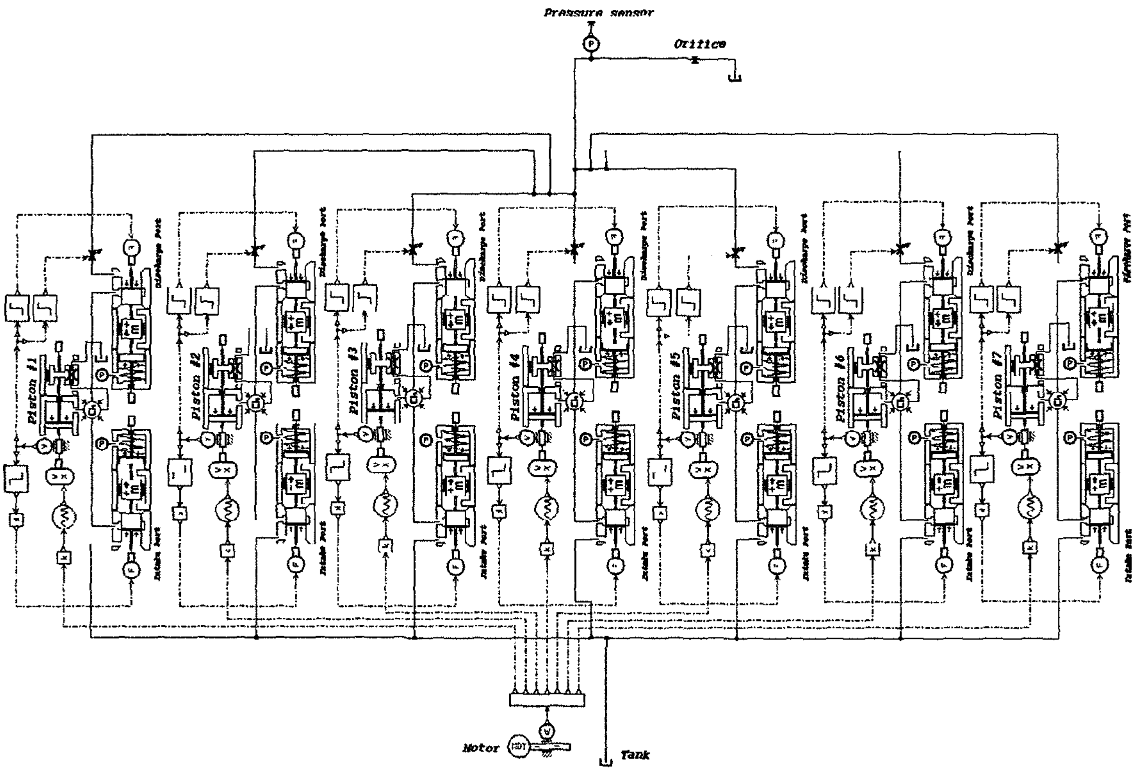

Figure 5. Modeling in AMESim environment.

pressure waves respectively.

## 3. MODELING

For the modeling of a bent-axis piston pump and hydraulic pipeline with a parallel line, AMESim (Imagine, S. A., 2000), which is a simulation environment oriented toward a global approach in hydraulic field, is used. In this study, a pump with 7 piston and a rated flow rate of 12 cc/rev was used. Figure 5 shows the modeling of the hydraulic system in AMESim environment.

## 4. EXPERIMENT

The pump is running at 1,000 rpm in this experiment. The discharge pressure is controlled by the orifice installed at the outlet of the hydraulic pipeline. The hydraulic pipeline is a steel pipe with an internal diameter of 10 mm, which is about 5,000 mm long from the inlet to the outlet. The direct length between the upstream junction and the downstream junction is 3,700 mm. The direct length of the parallel line is determined by Equation (19), and that of the lines before and after the parallel line part is 275 mm and 1,025 mm respectively. The oil is the type used for industrial hydraulic systems. Its density is 870 kg/m$^3$, the bulk modulus is $1.6 \times 10^9$ N/m$^2$ and the kinematic viscosity is $3.1 \times 10^{-5}$ m$^2$/s at 40°C.

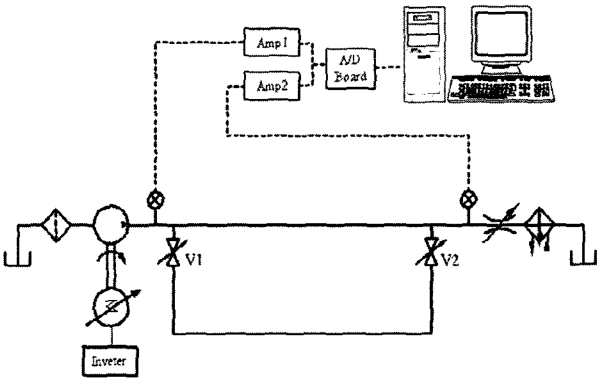

Figure 6. Schematic diagram of the experiment.

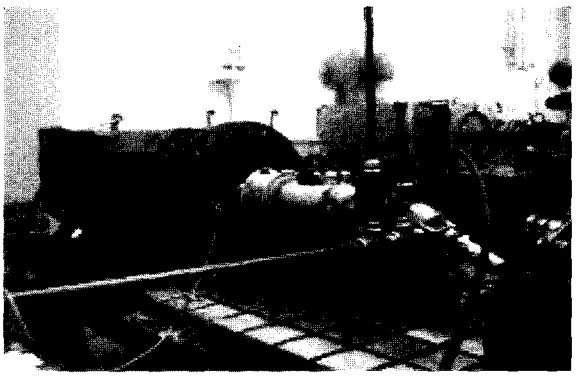

Figure 7. Photograph of the experimental apparatus.

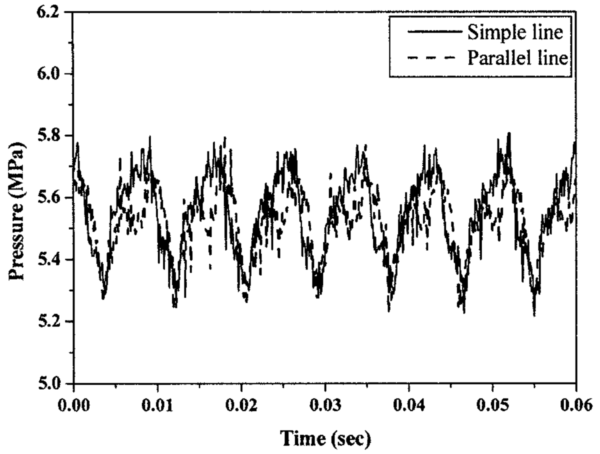

Figure 8. Comparison of Experimental results in inlet pressure between simple and parallel line.

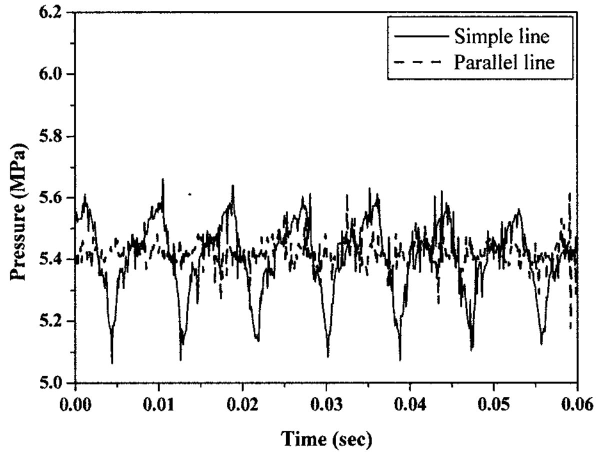

Figure 9. Comparison of simulated results in inlet pressure between simple and parallel line.

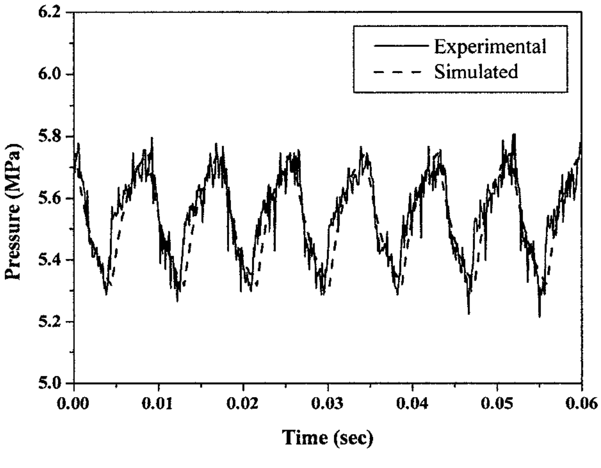

Figure 10. Comparison of experimental results in outlet pressure between simple and parallel line.

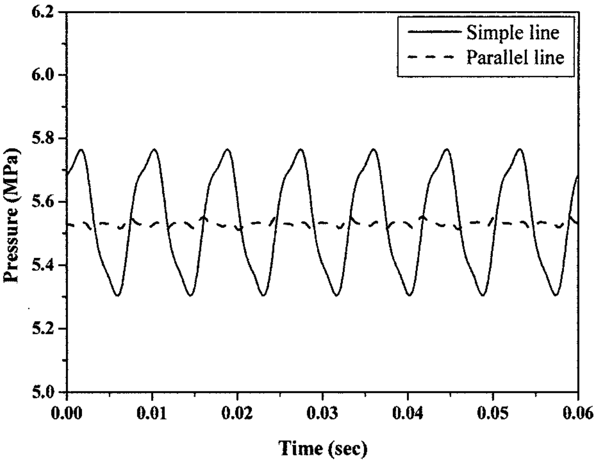

Figure 11. Comparison of simulated results in outlet pressure between simple and parallel line.

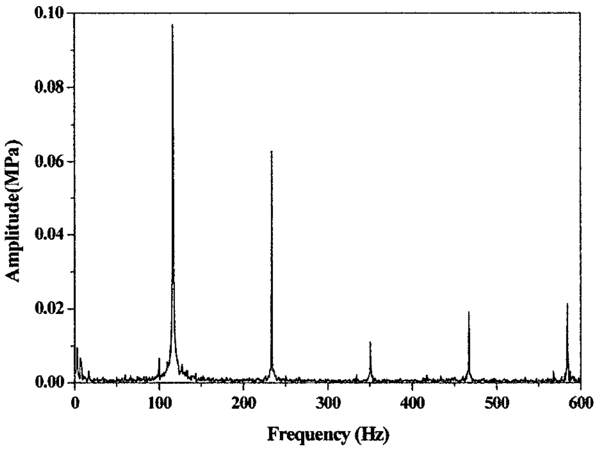

Figure 12. Frequency analysis of outlet pressure in simple hydraulic pipeline.

Pressures were measured at the inlet and the outlet of both the simple hydraulic pipeline with valves, V1 and V2, closed and the hydraulic pipeline with a parallel line with the valves V1 and V2, opened. The oil cooler is equipped to maintain the temperature of the working oil fixed at 40 ± 2°C. Figure 6 shows the schematic diagram of the experiment, and Figure 7 show the photograph of the experimental apparatus.

## 5. RESULTS

Figures 8, 9, 10 and 11 show a comparison between the experimental results and simulated results of pressure waves, which were measured at the inlet and outlet of the simple hydraulic pipeline and the hydraulic pipeline with a parallel line, respectively. From the comparison, we can observe that the hydraulic pipeline with a parallel line

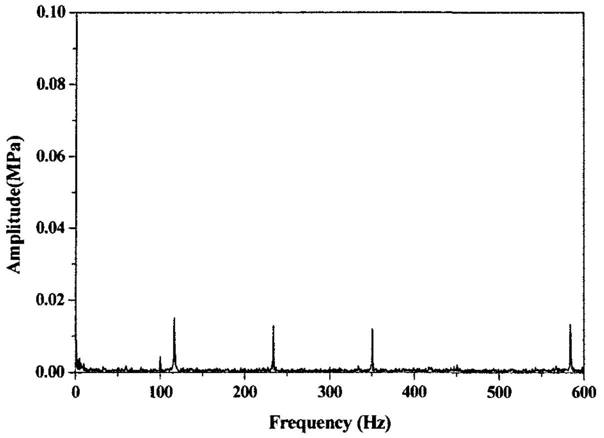

Figure 13. Frequency analysis of outlet pressure in hydraulic pipeline with a parallel.

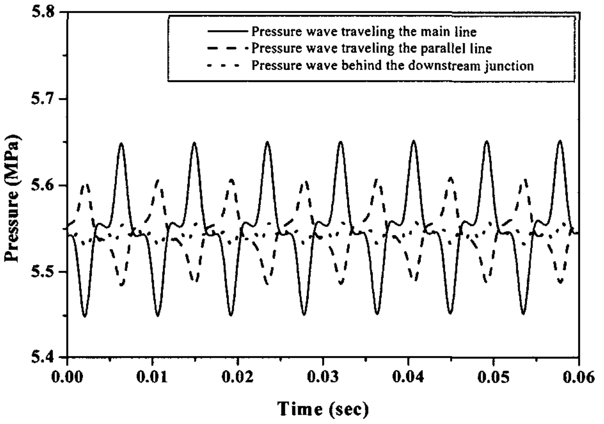

Figure 14. Effect of phase interference at downstream junction.

reduces the pressure ripple at the inlet as well as at the outlet. Especially, Figures 12 and 13 show that most of the pressure ripples at the outlet of the hydraulic pipeline with a parallel line was reduced completely. Also, the simulated results agree faily well with the experimental results.

From this fact, we can see that the parallel line in a hydraulic pipeline is effective in reducing pressure ripples. To compare the phase between the pressure wave traveling in the main line and the pressure wave traveling in the parallel line, we measured the pressure waves at the downstream junction of the hydraulic pipeline with a parallel line in the simulation. The results are shown in Figure 14. The phase between two pressure waves is almost 180° in both the experiment and simulation. This phase difference, known as the offset effect, is responsible for the reduction in the amount of pressure ripples.

## 6. CONCLUSIONS

In this paper, the ability of the parallel line to reduce the pressure ripples in hydraulic systems was investigated by experiment and simulation. The investigation gave the following results.

(1) The dynamic characteristics of the hydraulic pipeline with a parallel line are analyzed in the frequency domain.

(2) It is confirmed that the hydraulic pipeline with a parallel line can considerably reduce the amount of pressure ripples by using the phase difference of pressure waves caused by the length difference between the main line and the parallel line.

(3) The analysis model was developed for the hydraulic pipeline with a bent-axis piston pump, and the simulated results agree well with experimental results.

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