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Noise source identification and transmission path optimisation for noise reduction of an axial piston pump

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ABSTRACT

Reduction of the radiated noise of an axial piston pump by modifying the piston pump shell structure has previously been achieved. However, the main noise sources and transmission paths have scarcely been investigated. In this study, the flexible multi-body dynamic (FMBD) model of an axial piston pump was established, and the main noise source and transmission path were determined. Then, a set of programs to improve the sound quality of the piston pump was developed. Firstly, the finite element analysis method was adopted to numerically analyse the natural modes of the shell, which was verified by an experimental modal analysis using the multiple input and single output technology. Then, AMEsim and Virtual.lab were used to establish the FMBD model of the axial piston pump considering the bearing dynamics and fluid dynamics. The simulated vibration acceleration level from the shell surface was consistent with the experimental test. The finite element method (FEM) and Automatically Matched Layer (AML) technique were employed to acquire the radiated noise from the shell surface in the frequency range of interest, and the noise transmission path was analysed. From the results, it was determined that the excitation force of the swash plate to the cover Y direction is the main noise source. Finally, the main noise transmission path was optimised, and the optimised piston pump radiated noise is reduced by 1.26 dB(A) of the main noise frequency. This research can provide theoretical guidance for engineers to design low noise piston pumps.

1. Introduction

Axial piston pumps are widely used in hydraulic fluid power systems, especially in engineering machinery, because of their high rotation speed and high discharge pressure. However, their noise levels are higher than those of other displacement pumps. They are the main noise source in hydraulic systems. Previous studies have primarily focused on changing their structural parameters and shell shape to reduce radiation noise, but there are few detailed studies on noise generation and transmission paths of axial piston pumps.

The noise mentioned above is constituted of the fluid-borne noise and structural radiated noise. Flow ripple is the source of fluid-borne noise. It has been studied frequently by optimising structure parameters (slot length, width and height) of valve plate [1,2], addition the highly damped check valve [3] and variable reversing valves [4]. The fluid-borne noise is much lower than the structural radiated noise.

The structural radiated noise is mainly reduced by changing the structural parameters [5]. In axial piston pumps, early studies have demonstrated that the noise can be reduced by improving their stiffness. Palmen et al. [6]investigated on the noise reduction of an axial piston pump by means of structural modifications. Xu et al. [7] researched on noise reduction through modification of the housing structure, and the average sound pressure level was reduced by approximately 2 dB(A) at a discharge pressure of 250 bar. In their study, the research objective was to reduce the radiated noise of the piston pump; however, the cause of noise generation and the transmission paths were not analysed.

To accurately identify the noise sources and transmission paths, it is necessary to study the vibration performance of the piston pump, including a detailed identification of the different excitation sources, modal characteristics, and modelling of the forced vibration response of the pump assembly. Roccatello et al. [8]established a variable displacement axial piston pump model, but the dynamic excitation forces responsible for the vibration were not studied. Milind et al. [9] developed a combined multi-body dynamics finite element (MBD/FE) approach for modelling the dynamics and vibration behaviour, but the bearing dynamics model was not considered in the MBD / FE model. Even though the multi-body dynamics (MBD) model of the axial piston

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pump had previously been studied, their model was not complete and could not fully reflect the vibration performance of the piston pump.

The finite element method (FEM) and boundary element method (BEM) are widely used to calculate structural radiation noise. Guo et al. [10] predicted the radiated noise from the oil pan of a diesel engine by a coupling methodology of FEM and BEM. Mao et al. [11] investigated on the radiated noise of a four-cylinder diesel engine using the coupling methodology of FMBD and BEM, and the radiated noise was reduced by optimising the shell structure. Tang et al. [12] developed the fluid radiated noise for a variable displacement external gear pump by CFD and LAA methods. However, the radiated noise for a piston pump by the coupling methodology of FMBD and FEM-AML has rarely been studied in the currently available literature.

In the present work, the radiated noise source and transmission paths of an axial piston pump are studied. A coupling method for the radiated noise analysis of the piston pump, the so-called FMBD–FEM/AML, is proposed. The aim of this study includes the following aspects. (a) The finite element model of the piston pump shell for the modal analysis is built and experimentally validated. (b) a detailed FMBD model of the piston pump assembly is established by applying Virtual.lab and AMEsim. The FMBD model, including internal components, bearings, and the shell; and the fluid dynamics of the piston pump are taken into account. All the excitation forces transmitted to the cover and the housing are analysed, and the vibration performance of the piston pump is verified using experiments. (c) The FEM/AML method is adopted to calculate the radiated noise of the piston pump. The main noise source and noise transmission path are determined. (d) The cover of the piston pump is optimised by increasing the number of ribs, and the sound pressure level (SPL) of the maximum radiated noise frequency is reduced by 1.26 dB (A) at discharge pressure 300 bar.

The work conducted in this paper can be instructive to engineers for structural design and radiated noise analysis of the axial piston pumps. The flow chart of the radiated noise improvement design for an axial piston pump is shown in Fig. 1.

2. The piston pump shell natural modal analysis

The axial piston pump in the study has nine pistons. The rated pressure of the pump is 300 bar, the displacement is 112 cm^3/r, and the maximum speed is 1900 r/min. The finite element model of the shell is built. Subsequently, the shell natural mode is analysed based on the finite element model, and an experimental modal analysis is performed for the purpose of validating and updating the finite element model of the piston pump shell.

2.1. Finite element modal analysis (FEMA)

The shell of the piston pump is the ultimate receiver for vibration transmission, and is the body of radiation noise. The shell includes the housing and cover. It is necessary to analyse the natural modes because of the close relationship between radiated noise and natural frequencies. The FEM of the housing and cover are shown in Fig. 2. It consists of 100863 tetrahedral elements. Between the cover and housing is connected by four bolts. The bolts are modelled by 1D beam elements. The housing and cover are made of malleable cast iron with the density, elastic modulus and Poisson's ratio of 7010 kg/m^3, 152Gpa and 0.274, respectively. The first eight natural modal frequencies of the shell from the FEM analysis are listed in Table 1.

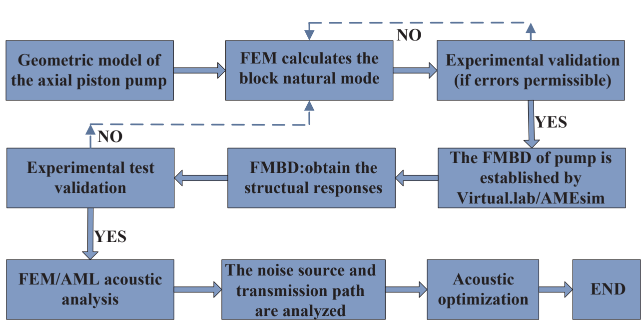

Fig. 1. The flow chart of the radiated sound improvement design for an axial piston pump.

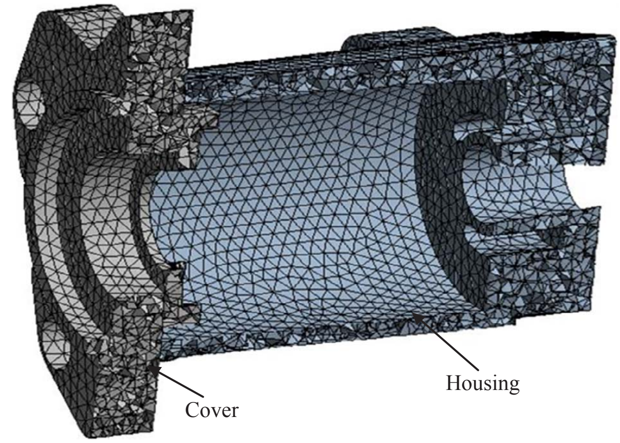

Fig. 2. The finite element model of the housing and cover.

2.2. Experimental validation of the FEMA

An experimental modal analysis (EMA) is performed using the LMS impact test. The schematic of the test rig is shown in Fig. 3. The shell is hung by a perfectly flexible rope, which has no influence on obtaining the natural modes. The EMA uses the multi-point percussion and single point response technology. There are 68 points (No. 1–68) constructed on the housing and cover surface. A three-way acceleration sensor (type 356A35,PCB) is connected to point 24. A hammer is moved to each measuring point to excite the housing and cover, and each measuring point is hit 3 times. LMS SCADAS is used for date acquisition, and the data analysis is performed in the LMS Test.lab modal analysis. The modal frequency and mode shape are obtained when the EMA is completed. The first eight natural modal frequencies of the housing and cover from the EMA are listed in Table 1. The maximum relative error is 4.31%, while other errors are smaller than 3%, as seen in Table 1. Thus, the error of the finite element model is permissible in engineering.

Further, to determine the accuracy of the finite element model, the FEMA and EMA shell shapes are shown in Fig. 4. The first mode shape is mainly the cover axial and housing horizontal shape. The second mode shape is mainly the cover and housing bending shape. The third mode shape is mainly the cover horizontal shape. The fourth mode shape is mainly the cover bending shape.

By analysing the above Table 1 and Fig. 4, the natural frequencies and shapes of the FEMA and EMA are basically consistent, which verifies the reliability of the finite element model. This ensures the accuracy of the numerical model of cover and housing, which will be

Table 1
Comparison of results from FEMA and EMA.

Order	FEMA (Hz)	EMA (Hz)	Error (%)
1	3102.562	3173.1	2.27
2	3549.922	3521.9	−0.78
3	4182.258	4206.8	0.58
4	4715.363	4918.6	4.31
5	5271.425	5401.5	2.47
6	5682.381	5639.6	−0.75
7	5736.632	5842.5	1.8
8	6760.294	6786.1	0.38