Applied Acoustics 110 (2016) 43–52

Contents lists available at ScienceDirect

Applied Acoustics

journal homepage: www.elsevier.com/locate/apacoust

Technical note

Numerical and experimental studies on housing optimization for noise reduction of an axial piston pump

Bing Xu, Shaogan Ye, Junhui Zhang *

The State Key Laboratory of Fluid Power and Mechatronic Systems, Zhejiang University, Hangzhou 310027, China

ARTICLE INFO

Article history:
Received 5 November 2015
Received in revised form 28 January 2016
Accepted 21 March 2016
Available online 26 March 2016

Keywords:
Axial piston pump
Noise reduction
Finite element method
Model updating
Topology optimization
Experimental measurement

ABSTRACT

This paper presents a methodology to reduce the noise of an axial piston pump through modification of the housing structure, combined with both numerical and experimental methods. The finite element models of the housing and cover are established, and are assembled together. The finite element models are validated and updated using experimental modal analysis. The frequency response function of the assembly is calculated, and the shell element in the inner surfaces of the housing is added. The effects of the thickness of the shell element on the frequency response function are identified. A topology optimization is conducted for the purpose of reducing the frequency response function and the increase of mass. The prototype pump is manufactured and assembled. Different experimental measurements are carried out, including the measurement of the vibration and the distributions of the sound pressure levels around the pump. Results show that the vibration and noise are reduced by using the optimized housing. In particular, the average sound pressure level is reduced by about 2 dB(A) at the discharge pressure of 250 bar, and the sound pressure level at the second harmonic is reduced significantly. The method proposed here can also be used for other kinds of displacement pumps.

1. Introduction

Axial piston pumps are widely used in hydraulic fluid power systems due to their ability of adjusting their displacements to match the actual demands. However, their noise levels are higher than other kinds of displacement pumps, and they are the main noise sources in hydraulic systems. In order to meet the requirement of stricter legal rules, the noise from hydraulic pumps must be reduced [1], particularly the axial piston pumps. A methodology was proposed to model the vibro-acoustic characteristics of a gear pump [1], and it is helpful in reducing the noise of hydraulic pumps.

The noise generated by an axial piston pump is divided into fluid-borne noise, structure-borne noise and air borne noise. Flow ripple is the source of fluid-borne noise; it causes the vibration of the hoses, valves and actuators when flow ripple interacts with system impedances, and a method is proposed to measure the pressure ripple of a vane pump [2]. Internal excitation forces and moments are the sources of structure-borne noise; they cause the vibration of the pump. Valve plate is the key part affecting the generation of noise sources [3], and the inertia effects were considered in modeling the pumping dynamics. Valve plates with different kinds of slot geometries were compared [4], demonstrating that the valve plate with triangular slot geometry is less sensitive to working conditions. Structure parameters (slot length, width and height) of this kind of valve plate are optimized to reduce the flow ripple based on a pump model [5,6]. This method cannot consider fluid-borne noise and structure-borne noise sources simultaneously. In order to account for both fluid-borne noise and structure-borne noise, a genetic optimization method was used [7]. Similar optimization method was employed to find the optimal cross-angle for reducing the sensitivity of noise to displacement [8]. In addition to the passive methods used, several active methods are also used to control the fluid-borne noise and structure-borne noise sources, among which, the highly damped check valve [9] and variable reversing valves [10] are found to be effective at different working conditions. These investigations focus on reducing the noise sources.

The other method to reduce the noise is to change the transmission paths [11]. For axial piston pumps, early studies have demonstrated that the noise can be reduced by adding reinforce ribs [12,13], changing the mounting and the shape of housing [14]. However, these modifications are carried out more or less by trial and error method. Even though the vibro-acoustic models of axial piston pumps were established, these models are merely used to

* Corresponding author.
E-mail addresses: bxu@zju.edu.cn (B. Xu), shaoganye@zju.edu.cn (S. Ye), benzjh@zju.edu.cn (J. Zhang).

http://dx.doi.org/10.1016/j.apacoust.2016.03.022
0003-682X/

Nomenclature

n	the rotational speed of the test pump
L_p	the sound pressure level
MAC_{ij}	the modal assurance criterion between the measured and simulated modes
p_0	the reference sound pressure, $p_0 = 1 \times 10^{-6}$ Pa
i, j	the ith mode of the measured modes and the jth mode of the simulated modes
ε	the displacement setting of the test pump
ψ_j^{FE}	the finite element modal vectors of mode j
$\{\psi_j^{FE}\}^*$	the complex conjugate of the finite element modal vectors of mode j
ψ_i^{test}	the measured modal vectors of mode i
$\{\psi_j^{test}\}^*$	the complex conjugate of the measured modal vectors of mode j

evaluate their vibration and noise at the design stage. It could hardly provide information on how to modify the housing.

Besides, comparing to the automotive industry, the development of quieter components falls behind in fluid power industry. In the automotive industry, the topology, size, shape and topography optimization methods are already employed to find the optimal structure for obtaining lighter and quieter machines [15,16]. It is expected that the fluid power industry could use these techniques to promote its advancement.

The aim of this study includes: (1) build a finite element models of the housing and cover for the modal analysis; (2) validate and update the finite element model using experimental modal analysis based on modal assurance criterion; (3) develop a topology optimization methodology to optimize the housing structure for the purpose of reducing the frequency response function; (4) evaluate the differences of vibration and noise between the original and optimized pumps using different experimental measurements, including vibration measurement and sound pressure levels measurement around the pump.

2. Finite element analysis

The axial piston pump is of variable displacement swash plate type with nine pistons. The rated pressure of the pump is 350 bar, the displacement is 250 cm^3/r, and the rated rotational speed is 1500 r/min. In this section, the finite element models of the housing and cover are built, and they are assembled together. The finite element modal analysis is performed, and experimental modal analysis is performed for the purpose of validating and updating the finite element model. The updated model will be used for the calculation of the frequency response function in Section 3.

2.1. Finite element modal analysis

The housing and cover in the axial piston pump are the main parts affecting the vibration transmission and noise emission. Their finite element models are established for performing the finite element modal analysis (FEMA). There are many lubrication interfaces in the axial piston pump. Among these interfaces, the slipper/swash-plate interfaces, piston/cylinder interfaces and cylinder/valve-plate interface are the most dominant. Even though different method is used to modelled these interfaces for the finite element analysis, the results are not good, making it difficult to perform the analysis correctly [17]. For these reasons, these parts are not included. This is acceptable not only for reducing the computing time, but also for identifying the real effects of the housing on the vibration and noise of the axial piston pump.

The finite element model is shown in Fig. 1. The tetrahedral element is used to model the housing and cover, there are totally 139,432 (92,030 + 47,402) tetrahedral elements. The bolts between the housing and cover are modelled by 1D beam elements having the same radius with the actual bolts (M18). The beam elements are connected to the surrounding elements using rigid spider element (rbe2) with three translational degree of freedoms (Dofs) constrained. The housing and cover are made of malleable cast iron. The recommended material properties are: $\rho = 7010$ kg/m^3, $\mu = 0.274$, $E = 1.61 \times 10^{11}$ N/m^2. In the initial stage, these properties are used to determine the modal characteristics of the housing and cover. The simulated first ten modal frequencies of the housing using these initial settings are listed in Table 1. Then, modal assurance criterion analysis (MACA) will be conducted to analyze the correction between the simulated and measured modal Eigen values. Later, These properties will be updated in order to improve the correction. When the properties are successfully updated, the frequency response function analysis will be performed. Their results are used in the topology optimization as the constraints as it will be presented in Section 3.

2.2. Experimental modal analysis

Experimental modal analysis (EMA) is performed using impact test. The schematic of the test rig is shown in Fig. 2. There are 74 points (No. 1–74) constructed on the housing surface, with the piezoelectric accelerometer (type 4507-B-001, Brüel & Kjær) attached to point 59. A hammer is moved to each measuring point to excite the housing, and the integrated inner force transducer in the impact hammer provides a measured input excitation. The accelerometer attached to the pump measured the acceleration, and the accelerometer provides the dynamic response. The measured input and output yield the system transfer functions. The modal frequency and mode shape are obtained when the EMA is completed.

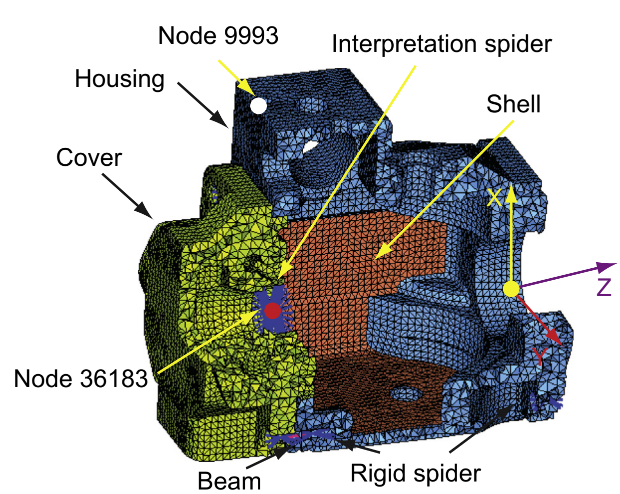

Fig. 1. The finite element model.