

Multi-objective Optimization Tool for Noise Reduction in Axial Piston Machines

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ABSTRACT

Noise generation in axial piston machines can be attributed to two main sources; fluid borne and structure borne. Any attempt towards noise reduction in axial piston machines should focus on simultaneous reduction of these two sources. A multi-parameter multi-objective optimization approach to design valve plates to reduce both sources of noise for pumps which operate in a wide range of operating conditions has been detailed in a previous work (Seeniraj and Ivantysynova, 2008). The focus of this paper is to explain the background and to demonstrate the functionality and usefulness of the methodology for pump design.

INTRODUCTION

Noise reduction in axial piston pumps (Fig. 1) has been attempted by many researchers with different design approaches and techniques. Most of the reductions techniques available today focus either on fluid borne noise (flow ripple) or structure borne noise (oscillating forces on swash plate). For overall reduction in noise, the effect of the reduction technique on both fluid and structure borne sources should be analyzed simultaneously. The performance and noise generation characteristics of any axial piston machine are dependent on a large number of variables. The contribution of each variable to the performance varies widely as the operating conditions vary. Such a multi variable system demands a lot of computational help to systematically investigate the parameters that have major influence on the noise sources. This paper discusses the approach of how noise generation at the source can be reduced and the overall performance improved in a systematic way with the help of computer based simulation and optimization.

STATE OF THE ART IN NOISE REDUCTION USING VALVE PLATE MODIFICATION

Over the past four decades, noise reduction in axial piston pumps has been addressed through several reduction techniques by researcher all over the world. A summary of reduction techniques available to reduce noise sources in axial piston and the advantages and

limitations of such techniques can be found in Seeniraj and Ivantysynova (2006a). Providing relief grooves at the end of kidney ports is one of the oldest noise reduction techniques in axial piston pumps. Relief grooves (Fig. 3) aid in gradual equalization of displacement chamber pressure to port pressure during pre-compression and decompression; thereby avoid huge reverse flows in or out of the displacement chamber. Several works detail the effect of relief grooves on flow ripple (Helgestad (1974), Zeiger and Akers (1985), Edge and Darling (1986, 1989), Palmberg (1989) and several others).

It is known that, a valve plate with relief grooves is a passive reduction technique designed for a particular operating condition and does not guarantee noise reduction for other operating conditions. But properly designed grooves can effectively function over a wide range of operating conditions (Taylor 1980, Johansson 2005). In this paper, optimization of relief grooves for a pump working in a range of operating conditions is explained.

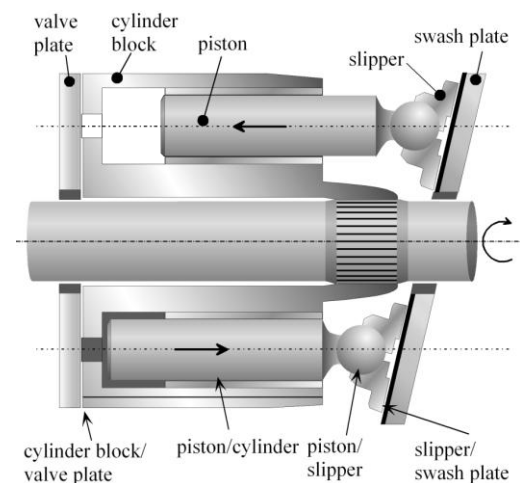


Fig. 1 Schematic of an axial piston swash plate pump

COMPUTER AIDED VALVE PLATE OPTIMIZATION

Traditional valve plate design involves progressively evaluating several valve plate designs by varying certain

valve plate parameter and studying the effect of the parameter on noise sources. But reducing fluid borne and structure borne noise sources simultaneously involves analyzing several parameters at a time and evaluating the effect of change in parameter on several objectives including volumetric efficiency. Several constraints on the optimization such as cavitation or over-pressurization inside displacement chamber need to be checked with the change in parameters. Also, the effect of FBN and SBN sources needs to be analyzed both in time and frequency domains. Even to an expert pump designer, it becomes an overwhelming task to achieve all of the mentioned things. There are several software tools available for simulation and analysis of axial piston pumps.

Even with the help of software tools, varying valve plate parameters one by one at a time in a particular direction; simulating each design using software tools; and studying the effect on noise sources; changing the parameter again takes a long time. Also it is hard to visualize if more than two parameters are affecting all the objectives. Manually varying the parameter was tried with partial success in a previous work (Seeniraj and Ivantysynova 2006b). It is learnt from the previous work that the manual process demands a lot of expertise on the part of the designer. Also manual optimization is limited by the designers' ability to track the effect of all the groove parameters on noise sources in a multi-objective domain. Thus it is important to analyze the parameters that affect noise generation in axial piston pumps systematically with the help of simulation tools. This task can be handled efficiently with Multiobjective Optimization Genetic Algorithms (MOGA). As genetic algorithms have evolved over the years, there are several methods within MOGA which can be adopted to solve the valve plate optimization problem.

SIMULATION MODEL

This section describes the mathematical model of axial piston pump which forms the core of multi-parameter optimization methodology. The goal of the model is to capture main physical quantities of interest which include *instantaneous displacement chamber pressure* (p), *flow pulsation in discharge* (Q_{rHP}) and *suction ports* (Q_{rLP}), *oscillating forces and moments* on the swash plate and *volumetric loss* due to compressibility and cross flow. A more in depth model of the pump can be found in Ivantysyn and Ivantysynova (2001).

Pressure (p) inside each displacement chamber at any instant is determined using a control volume approach (Fig. 2). Rate of change in pressure inside displacement chamber can be written from the definition of bulk modulus as:

$$\frac{dp}{dt} = -\frac{K}{V} \left(\frac{dV}{dt} - Q_r - Q_s \right) \quad (1)$$

where V is the fluid volume in displacement chamber expressed as:

$$V = V_0 - sA \quad (2)$$

V_0 is the volume of fluid in the chamber when the piston is at ODC and A is the cross sectional area of the piston. Rate of change in displacement chamber volume due to piston movement can be expressed as a function of velocity of piston (v) and piston area of cross section (A) as:

$$\frac{dV}{dt} = vA \quad (3)$$

Q_r is the flow rate in and out of displacement chamber. Using Eq. 2 and 3, Eq. 1 can be rewritten as:

$$\frac{dp}{dt} = -K \left(\frac{vA - Q_r - Q_s}{V_0 - sA} \right) \quad (4)$$

Q_s is the sum of gap flows between piston-cylinder block, piston-slipper and cylinder block -valve plate. In this work, the effect of gap flow has been neglected and hence not included in the investigation of different noise reduction techniques. The flow rate Q_r can be expressed using the orifice equation as,

$$Q_r = \alpha_D A_r \sqrt{\frac{2}{\rho}} \sqrt{p - p_r} \text{sgn}(p - p_r) \quad (5)$$

where p_r is the pressure at the pump port. A_r is the area of the smallest cross section of the flow passage between displacement chamber and pump ports (discharge and suction). A_r is determined by the profile of the valve plate openings (Ivantysynova 2004).

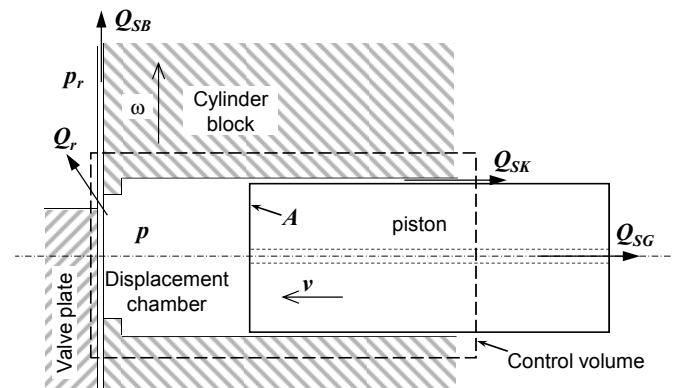


Fig. 2 Schematic of displacement chamber enclosed by piston, cylinder block and valve plate

Instantaneous chamber pressure is directly related to the forces imparted on the different components of the pump. Each piston exerts a force on the swash plate, termed as pressure force (F_p), proportional to the instantaneous displacement chamber pressure (p). Pressure force can be expressed as:

$$F_p = A(p - p_l) \quad (6)$$

where p_l is the case pressure. As p alternates between suction and discharge pressure, so does the force applied by each piston on the swash plate. Pressure force (F_p), is one among the three forces applied on the swash plate by the pistons. The other two, inertia force and friction force, are small compared to pressure force and their effect can be neglected for this study. Resultant pressure force on the swash plate is the sum of pressurized pistons. Swash plate moments M_{SX} , M_{SY}

and M_{SZ} generated due to the application of pressure forces can be expressed as:

$$M_{SX} = \frac{R}{\cos^2 \beta} \sum_{i=1}^z F_{pi} \cos \varphi_i \quad (7)$$

$$M_{SY} = R \sum_{i=1}^z F_{pi} \sin \varphi_i \quad (8)$$

$$M_{SZ} = -R \tan \beta \sum_{i=1}^z F_{pi} \sin \varphi_i \quad (9)$$

The output of the mathematical model includes time histories of pressure in displacement chamber, flow ripple at both ports, swash plate moment about x axis and evaluated objectives. The evaluated objectives include,

- amplitude of discharge flow ripple (ΔQ_{HP}),
- amplitude of suction flow ripple (ΔQ_{LP}),
- amplitude of oscillating moment on swash plate about swivel axis (ΔM_X),
- volumetric leakage (Q_{LK}) and
- mean of oscillating swash plate moment.

FORMULATION OF OBJECTIVE FUNCTION

As mentioned already, both FBN and SBN sources need to be minimized to have a quieter pump. It is important to note that any modification to the valve plate affects the amplitude as well as frequency of the noise sources. Thus it is necessary that the objective reflects both time and frequency domain (Johansson 2005, Ericson *et al.* 2008 among others). In this study, as there are several parameters to be considered along with four objectives, it is decided to keep the objective function simple with only time domain variables. Consecutively, the goal of optimization problem in this study is minimization of four objectives namely discharge flow pulsation (ΔQ_{HP}), suction flow pulsation (ΔQ_{LP}), amplitude of swash plate moment (ΔM_X) and volumetric loss (Q_{LK}). A multi-objective optimization (MO) of 4 objectives and n parameters can be formulated as follows:

$$\begin{aligned} &\text{Minimize } (\Delta Q_{HP}, \Delta Q_{LP}, \Delta M_X, Q_{LK}) = (\mathbf{x}) \\ &\text{where } \mathbf{x} = [x_1 \dots x_n]^T \\ &\text{s.t. } x_i^l \leq x_i \leq x_i^h \quad i = 1 \dots n \end{aligned} \quad (6)$$

with no cavitation and no over-pressurization

where $\mathbf{x}$ is a vector of valve plate parameters with each parameter having a range of values from x_i^l to x_i^h (Fonseca and Fleming 1993, Deb 1999, Johansson 2005). Valve plate parameters are discussed in the following section.

The optimization aims to achieve smooth precompression and decompression inside displacement chamber. Smooth compression means smaller rate of pressurization and depressurization which is directly related to the frequency content of the swash plate excitation. This is part of the reason why frequency

domain objectives are not included in the optimization problem.

It is known from prior investigations (Ivantysynova *et al.* 2005; Seeniraj and Ivantysynova 2006a, 2006b) that it is hard to find one particular valve plate design that will minimize all the objectives in the entire pump operating range. The expected solution to the optimization problem is a set of designs that are Pareto optimal rather than a single design. Though there are several approaches to a MO, the methodology adopted in this study can be classified under 'Multiobjective Optimization Genetic Algorithm (MOGA) with progressive articulation of preferences' (Fonseca and Fleming 1993). The decision maker (DM) is the pump designer who influences the methodology at several steps during evolution.

RELIEF GROOVE PARAMETERS

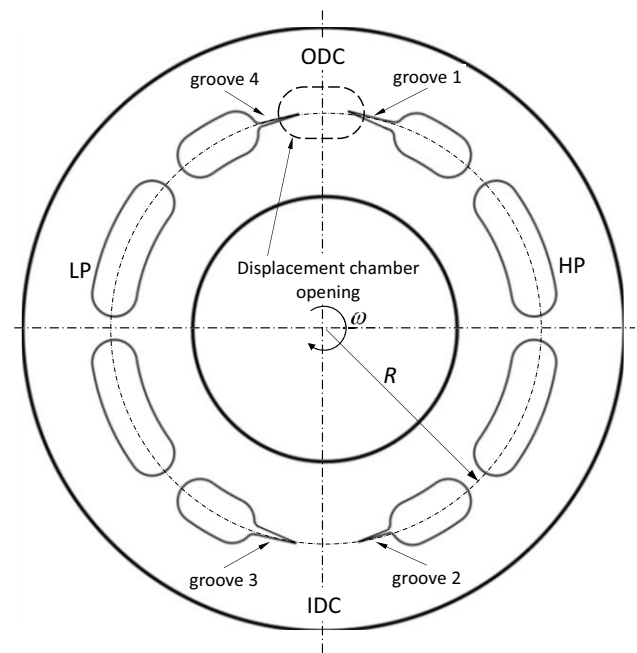


Fig. 3 Valve plate with grooves showing the displacement chamber opening

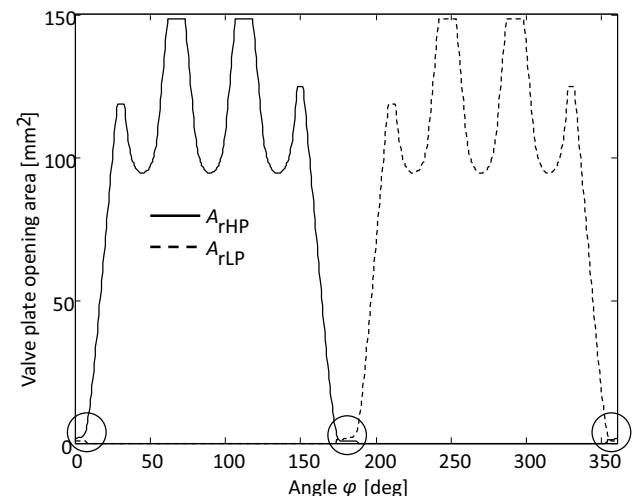


Fig. 4 Areas available for flow transfer between pump ports and displacement chamber (solid: discharge, dashed: suction)