A MULTI-PARAMETER MULTI-OBJECTIVE APPROACH TO REDUCE PUMP NOISE GENERATION

Ganesh Kumar Seeniraj and Monika Ivantysynova

Purdue University, Department of Agricultural and Biological Engineering, Maha Fluid Power Research Center, 1500 Kepner Drive, Lafayette, IN 47905, USA
gseenira@purdue.edu, mivantys@purdue.edu

Abstract

Noise emission from axial piston machines has been studied for several decades by many researchers and pump manufacturers. Different design methods for reducing the sources of pump noise have been proposed and are in use. The authors have studied and compared the effectiveness of several passive design methods. This paper presents a short overview of the existing design methods. The challenges in reducing both fluid borne noise sources (FBNS) and structure borne noise sources (SBNS) in a unified way are discussed. A computer aided multi-objective optimization procedure, which helps minimize the pump noise sources in a broad operating range, has been proposed by the authors. The optimization procedure is described in detail along with the mathematical model of the pump in this paper. An important contribution of the multi-objective parameterized approach is that the compression and the expansion region of the valve plate are simultaneously optimized unlike most previous works which consider compression and expansion separately. The parameterization of the valve plate is also explained. A case study and noise level measurements to prove the effectiveness of the optimization procedure are included.

Keywords: noise reduction, axial piston pump, multi objective optimization, precompression grooves, precompression filter volume

1 Introduction

A lot of passive and active design methods have been proposed earlier to reduce pump noise at the source. In the early 1970s, the majority of research on noise reduction in axial piston pumps (Fig. 1) focused on reducing the vibration of the pump casing, which was assumed to be the main structure borne noise source (SBNS) (Becker, 1970; Helgestad, 1974; Yamauchi and Yamamoto, 1979; Zeiger and Akers, 1985). Until the late 1980s, flow pulsations were not mentioned as a significant noise source. A major research project in hydraulic system noise reduction was started in 1976 in the United Kingdom and the project helped bring together different issues contributing to hydraulic system noise (Edge, 2000) and initiated several studies that led to the development of new design principles for noise reduction.

Different approaches have been studied over the years and they can be divided into two categories - (i) changing the pump design to reduce the noise at the source, (ii) blocking or canceling the noise transmitted from the pump. A comprehensive summary of noise reduction methods and their advantages and limitations can be found in the works Harrison (1997) and Johansson (2005). The most relevant methods that have been proposed and studied for axial piston machines are presented here.

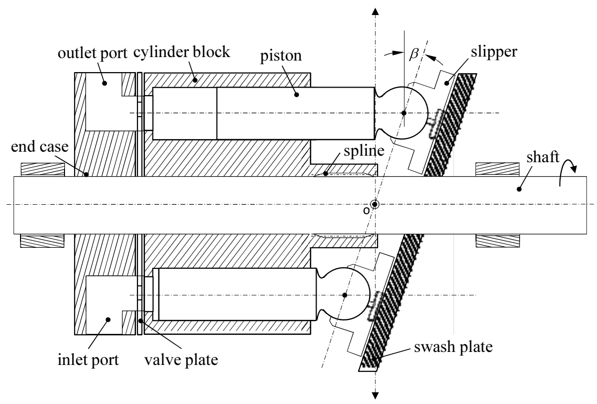

Fig. 1: *Schematic of an axial piston swash plate pump*

Ideal timing - The oldest technique namely *ideal timing* of valve plates achieves optimal compression by timing the opening of the ports (Helgestad, 1974; Yamauchi and Yamamoto, 1976; Edge, 1989; Pettersson, 1995). Ideal timing was effective only for a particular

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operating condition it is designed for and only for fixed displacement pumps. Ideal compression can be achieved by using relief check valve(s) that open(s) to discharge only after displacement chamber is fully pressurized (Helgestad et al., 1974). Weddfelt et al. (1994) proposed a vortex diode replacing the moving part of check valve and concluded that the response of the diode was not fast enough. A modified check valve termed heavily damped check valve (HDCV) overcame the limitations of relief check valves and performed well over a wide range of operating conditions (Harrison and Edge, 2000). The focus of the HDCV was to reduce FBN. Performance of the HDCV for speeds above 1500 rpm is not reported. Also swash plate instability was reported during the operation of the pump fitted with the HDCV (Harrison, 1997). The effect of HDCV on SBN needs further investigation. Implementation of HDCV is expensive and reduction in FBNS is not better than using precompression filter volume (Johansson, 2005). Becher and Helduser (2000) proposed a ring valve to reduce flow pulsations. The ring valve acts like a check valve and connects the displacement chamber to the discharge port depending on pressure differential. Results were evaluated only on discharge pressure ripple without mentioning SBNS.

Precompression grooves - The most common technique, using *precompression grooves*, is less sensitive to pressure levels and speeds than ideal timing. Precompression grooves spread out the compressibility effect (Petterson et al., 1991; Harrison, 1997). Precompression grooves achieve compression using high pressure fluid from the discharge port. Rate of compression is controlled by geometry of precompression grooves limiting back flow from the discharge port into the displacement chamber (Palmberg, 1989). Achieving compression with fluid from the discharge port creates a strong correlation between groove geometry and flow ripple. Also precompression grooves have influence on the volumetric efficiency if there is cross porting between discharge and suction ports (Pettersson et al., 1991).

Precompression filter volume (PCFV) represents a solution to weaken the correlation between compression and flow ripple, proposed by Petterson et al. (1991) and further investigated by Petterson (1995) and Johansson (2005). PCFV uses a volume of pressurized fluid attached behind the valve plate for pressurization of the displacement chamber. Jarchow (1997) investigated various possible configurations of connecting the PCFV in combination with check valve to reduce pressure pulsations at pump discharge; however the research entirely neglected rate of pressurization and SBNS. PCFV has a better FBNS reduction potential in a wide range of operating conditions over most other techniques (Johansson, 2005). However, the discussion on the size of the PCFV presented by Johansson (2005) is inconclusive. Though reduction in FBNS is satisfactory when using a PCFV, the rate of compression is increased which results in increase of SBNS (Ivantysynova et al., 2005). The sizing of PCFV is explained with calculations in Seeniraj (2009).

Active methods - Johansson et al. (2002) proposed a design named cross-angle that provided an additional adjustable inclination to the swash plate to achieve optimum compression for a wide range of operating conditions. Experimental investigation shows that cross-angle design is noisier than precompression grooves due to higher rate of pressurization (Johansson, 2005). Weingart (2004) investigated the possibility of an expanding stack of piezoelectric actuator to supply extra flow required during precompression to reduce flow pulsation. Piezoelectric material currently available cannot expand large enough to supply the required flow. Additional control is needed to synchronize expansion of piezoelectric material with variation in operating condition or else rate of precompression will be adversely affected. Also, there is a lot of research reported in reducing FBNS using cancellation devices that are attached to the pump discharge (Ortwig, 2005 among others). Such devices either cancel or reduce flow pulsation by supplying additional flow to fill the compressibility loss or acting as filters to absorb and prevent pulsations from being transmitted with or without additional control effort.

Prior work by the authors has shown that, of the available passive design methods (summarized in the previous passages) *precompression grooves* and *PCFV* have the possibility of minimizing the noise sources (FBNS and SBNS) in a range of operating conditions. These two methods and a combination of these two methods were further investigated by the authors. This paper details the investigations on precompression grooves and the multi-parameter multi-objective optimization methodology used to minimize the pump noise sources using precompression grooves.

For a given pump geometry and finite number of pistons, the design of the rotating group has the most effect on the flow pulsations in discharge/suction ports (fluid borne noise source, FBNS) and oscillating forces/moments on the swash plate (structure borne noise source, SBNS). There are other sources of vibrations external to the rotating group such as the prime mover and shaft bearings, and vibrations induced due to pressure ripple in system components such as relief valves and throttling orifices also contribute to overall system noise. In this work, the focus is to minimize both FBNS and SBNS by the proper design of the rotating group, especially the valve plate. Consequently, the physical quantities that are affected by the rotating group design are,

- amplitude of discharge flow ripple (ΔQ_{HP})
- amplitude of suction flow ripple (ΔQ_{LP})
- amplitude of oscillating forces on swash plate along three axes (ΔM_{X}, ΔM_{Y}, ΔM_{Z})
- volumetric efficiency/leakage flow rate (Q_{LK})

2 Challenges on Valve Plate Optimization

The purpose of the valve plate is to facilitate connection of individual displacement chambers to the suction and to the discharge port. The design of the valve plate openings can be used to determine the starting point (angular position) and the rise of pressure (p) in the displacement chamber from suction pressure (p_{s})

to discharge pressure (p_d) during *compression* and the drop in displacement chamber pressure from p_d to p_s during *expansion* (Fig. 2).

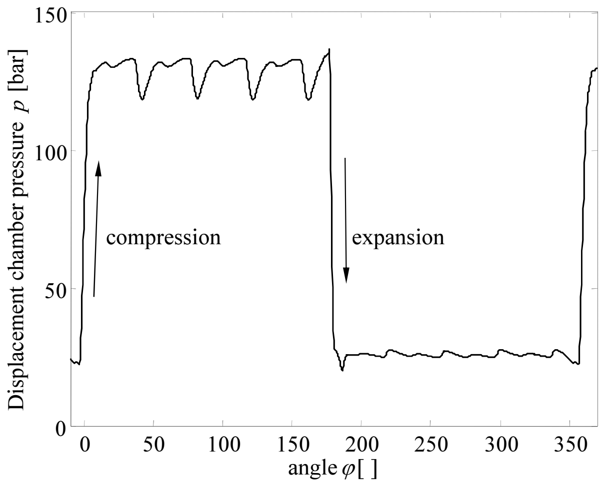

Fig. 2: *Plot of simulated displacement chamber pressure over one shaft revolution*

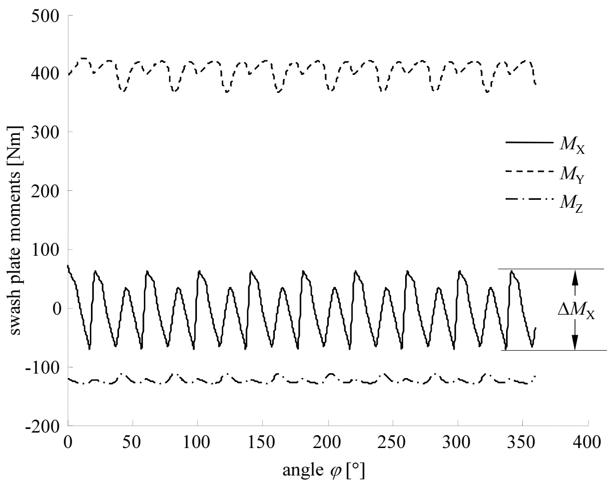

Fig. 3: *Plot of simulated swash plate moments over one shaft revolution of the pump*

A well designed valve plate should satisfy two conditions during compression.

1 Compression should take place to allow for minimum flow ripple at discharge. If compression is not enough ($p < p_d$), a back flow occurs from discharge into the displacement chamber which increases the amplitude of flow ripple. If compression is over done ($p > p_d$), a pressure peak appears inside the displacement chamber. The pressure peak increases the amplitude of the swash plate moment resulting in higher SBNS. Over-pressurization will also introduce a flow peak in discharge flow rate which finally results in higher flow ripple amplitude.

2 Compression should be smooth -i.e. the rate of compression (dp/dt) should be as small as possible. The rate of compression is directly related to forces applied on the swash plate. The magnitude of resultant pressure force (ΣF_{pi}) on swash plate, at any instant, depends on the pressure in each of pressurized cylinders. As pistons enter and leave the high pressure port, the magnitude of the resultant force switches between a maximum and minimum. The rate of switching from minimum to maximum depends on how fast a piston is pressurized and hence directly related to the rate of compression. Fig. 3 shows the plot of M_{SX} for one revolution of pump rotation for a 9 piston pump.

The same analogy in conditions can be applied to expansion.

1 If expansion is not enough ($p < p_s$), the high pressure fluid still trapped inside the displacement chamber will be discharged rapidly into suction port, increasing the amplitude of suction flow pulsation. If the displacement chamber is not opened to suction soon enough, the pressure in the displacement chamber could fall below the vapor pressure resulting in cavitation.

2 The rate of expansion should also be as small as possible for same reasons as the rate of compression.

3 Computer Aided Multi-objective Optimization

Traditional valve plate design involves progressively evaluating several valve plate designs by varying certain valve plate parameter and studying the effect of that 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 multiple objectives. Also constraints such as cavitation or over-pressurization inside displacement chamber need to be checked with the change in parameters.

There are several software tools available for simulation and analysis of axial piston pumps. Even with the help of the software tools, varying valve plate parameters one by one at a time in a particular direction and studying its effect on noise sources takes a long time. Also it is hard to visualize if more than two parameters are affecting all the objectives. Manually varying each of the valve plate parameters was tried with partial success by the author previously (Seeniraj and Ivantysynova, 2006). It is inferred from the previous work that the manual process demands a lot of expertise and effort on the part of the designer. Also manual optimization is limited by the designers' ability to track the effect of all the parameters on noise sources in a multi-objective domain. Thus it is important to analyze the parameters that affect noise sources in axial piston pumps systematically with the help of simulation based optimization tool. Also, in the previous work, manually assisted optimization was carried out without adhering to any particular optimization algorithm. But as the research progressed, it was intuitive to look for an optimization algorithm that would closely resemble the manually assisted optimization procedure. Consequently, *OptimVP*, a computer assisted optimization procedure based on the Multi-Objective Genetic Algorithm (MOGA) has been proposed for the valve plate optimization.

Literature is available on the application of MOGA for general engineering systems (Fonseca and Fleming, 1993; Deb, 1999; Andersson, 2001 among others). But there are not many works available on adopting the MOGA to noise source reduction for axial displacement machines. Johansson (2005) is the only relevant work which applies the MOGA approach to reduction of noise sources in axial piston pump. Johansson (2005) mentions optimizing three different design methods (ideal timing, precompression grooves and PCFV) at a single operating condition and presents a comparison of objectives between these three different reduction methods. But the details of design parameters such groove locations, the range of parameters analyzed, the location of PCFV and the methodology of arriving at an optimal design are not presented.

The optimization procedure proposed in this work uses a multi-objective optimization of 6 objectives (representing FBNS and SBNS) and n parameters. The optimization problem can be formulated as:

Minimize

$$(\Delta Q_{\mathrm{HP}}, \Delta Q_{\mathrm{LP}}, \Delta M_{\mathrm{X}}, \Delta M_{\mathrm{Y}}, \Delta M_{\mathrm{Z}}, Q_{\mathrm{LK}}) = (\boldsymbol{x}) \qquad (1)$$

where $\boldsymbol{x} = [x_1 \ldots x_n]^{\mathrm{T}}$

$$\text{s.t. } x_{\mathrm{i}}' \le x_{\mathrm{i}} \le x_{\mathrm{i}}^{\mathrm{h}} \qquad \mathrm{I} = 1\ldots..\mathrm{n}$$

with no cavitation and no over-pressurization where $\boldsymbol{x}$ is a vector of valve plate parameters (detailed in the next section) with each parameter having a range of values from x^{l} to x^{h} (Seeniraj, 2009).

3.1 Valve Plate Parameters

Figure 4, shows a typical valve plate with four precompression grooves. Figures 5 and 6 show the plot of the smallest possible areas available for flow transfer between displacement chamber and pump port for one full rotation of pump cylinder block. Of the areas shown in Fig. 5, only a small portion of areas (circled in Fig. 5) near piston dead centres is generated through the precompression grooves. Fig. 6 shows the zoom of the circled regions in Fig. 5.

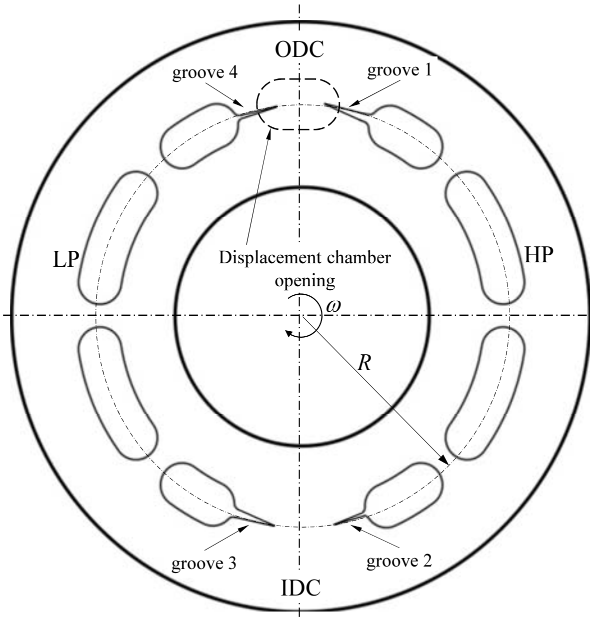

Fig. 4: *Valve plate with grooves showing the displacement chamber opening*

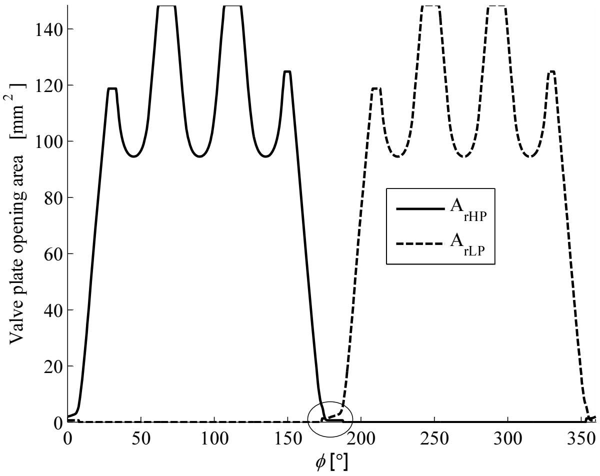

Fig. 5: *Areas available for flow transfer between pump ports and displacement chamber (solid-discharge; dashed-suction)*

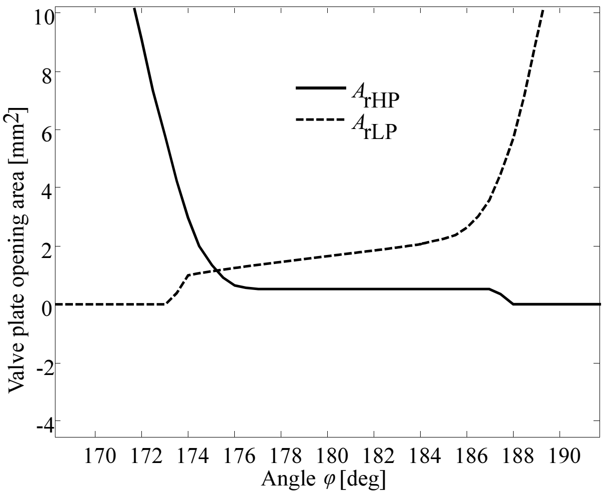

Fig. 6: *Enlarged section of areas available for flow transfer between pump ports and displacement chamber (solid-discharge; dashed-suction)*

Figure 7 shows a portion of valve plate near outer dead center (ODC) with displacement chamber at two different positions (I and II). At position I, displacement chamber is about to open to groove 1 leading to discharge port. At position II, displacement chamber is at the end of groove 1. The angle between the centre of displacement chamber opening at position I and the ODC axis is defined as φ_{s1}. Similarly, the angle between centre of displacement chamber at position II and ODC axis is defined as φ_{e1}. In Fig. 7, bold lines on valve plate shows precompression groove opening to the high pressure (referred as groove 1) and the corresponding area open to the displacement chamber is shown. Angles φ_{s1} and φ_{e1} represent opening and closing location of groove 1 and m_1 represents the slope of opening area controlled by groove 1. Figure 7 also shows groove 4, which is located at the end of suction port kidney. Thus area controlled by each precompression groove can be decomposed into three parameters – starting location (φ_{s}), ending location (φ_{e}) and slope (m). There are 4 precompression grooves in a typical valve plate making a total of 12 parameters (refer Table 1) and these parameter represent vector $\boldsymbol{x}$ in Eq. 1. It is necessary to mention here that the 12 parameters represent the smallest groove opening areas available for flow transfer between displacement chamber and pump ports. The groove parameters only represent the areas affect by the grooves (circled in Fig. 5). The rest of the opening area is taken from any typical valve plate that is initially available at the start of the optimization.

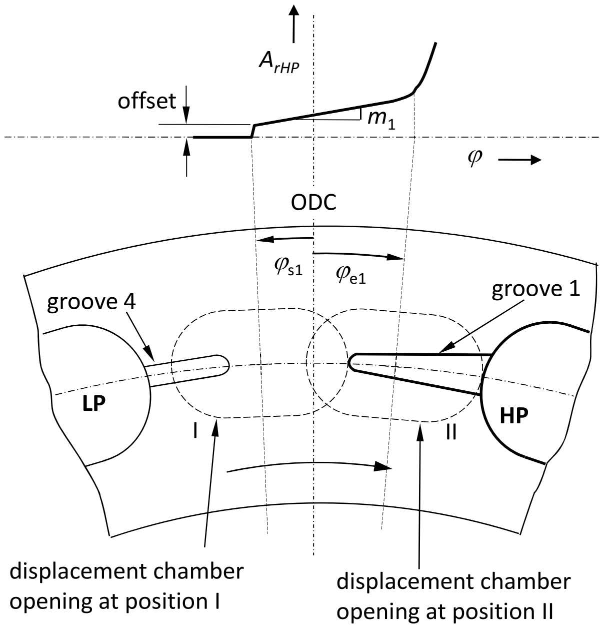

Fig. 7: *Part of valve plate near ODC showing displacement chamber at the start*

Table 1: List of precompression groove parameters

Precompression groove parameters	Unit
Starting location of groove 1 (φ_{s1})	[°]
Ending location of groove 1 (φ_{e1})	[°]
Starting location of groove 2 (φ_{s2})	[°]
Ending location of groove 2 (φ_{e2})	[°]
Starting location of groove 3 (φ_{s3})	[°]
Ending location of groove 3 (φ_{e3})	[°]
Starting location of groove 4 (φ_{s4})	[°]
Ending location of groove 4 (φ_{e4})	[°]
Slope of groove 1 (m_1)	[mm^2/°]
Slope of groove 2 (m_2)	[mm^2/°]
Slope of groove 3 (m_3)	[mm^2/°]
Slope of groove 4 (m_4)	[mm^2/°]

The valve plate parameters do not represent the actual geometry of the grooves, rather they define the areas available for flow transfer between the displacement chamber and the ports with respect to ODC. Different groove geometries can create the same opening area. In other words, a particular groove opening area can be achieved in several ways using different groove geometries, valve plate thicknesses, displacement chamber opening profiles and manufacturing processes. It is not the focus of this study to investigate how to manufacture a certain opening area on a valve plate.

3.2 Optimization Procedure

The purpose of the optimization procedure is to support pump designer with a computer-aided methodology in finding a valve plate opening area parameters that would enable a quieter operation of the pump in a chosen operating range. The proposed optimization procedure is shown as a flow chart in Fig. 8.

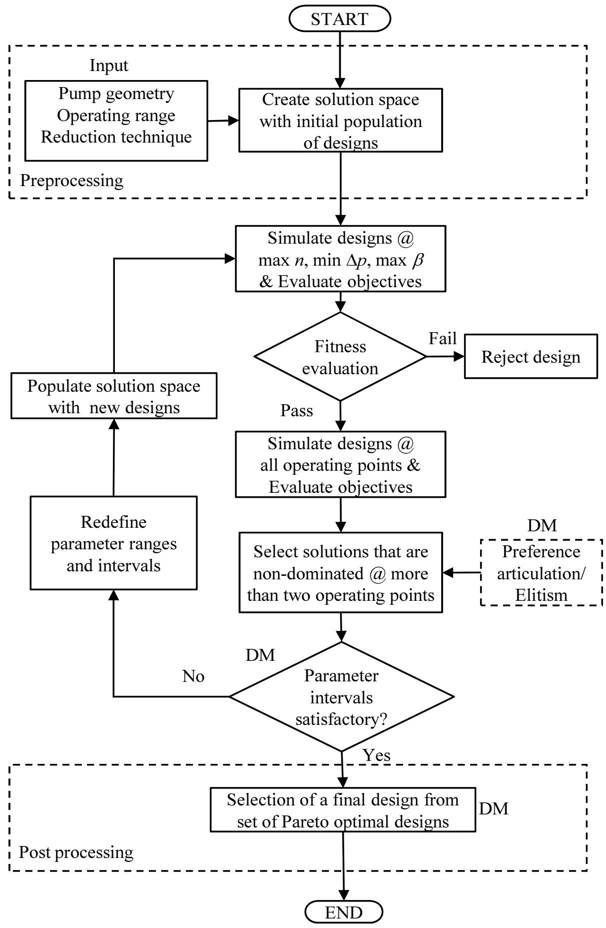

Fig. 8: *MOGA schematic*

Preprocessing includes collecting inputs specific to the given pump such as pump geometry, operating range (operating pressure, rotational speed and pump displacement) and valve plate geometry. Preprocessing also includes assigning initial range of values to each of the 12 parameters and creating a solution space with initial designs/population. Each design in the solution space is a possible solution to the optimization problem. Hence designs are also referred to as solutions. Initial designs are obtained by factorial combination of all the parameters with each parameter (x_i) allowed to vary from lower (x_i^l) to upper (x_i^h) limit. Prior investigations (Ivantysynova et al., 2005; Seeniraj and Ivantysynova, 2006) on manual optimization of grooves by the author helped identifying possible initial ranges for all parameters. Without prior knowledge, the initial value of parameters would fall under a wide range. This would increase the computational effort and time required to reach the final solution. The size of solution space also depends on the chosen range and the selected interval for each parameter. Choosing finer intervals for each parameter considerably increases the number of initial designs but it enables reaching the optimal solution in fewer searches and the inverse holds true.

All designs in the solution space are simulated using a mathematical model of the pump. The goal of the model is to capture main physical quantities of interest which include the displacement chamber pressure (p), flow pulsation in the discharge (ΔQ_{HP}) and suction (ΔQ_{LP}) ports, amplitude of the swash plate moments about the coordinate axes (ΔM_X, ΔM_Y and ΔM_Z), volumetric loss (ΔQ_{LK}) due to compressibility and cross flow.

The mathematical model of the axial piston pump used in the heart of the optimization procedure is detailed here.

The pump has z displacement chambers. Each displacement chamber is simulated and their effects summed up to simulated the entire pump. The rate of change in pressure inside each displacement chamber (dp/dt) can be calculated using a control volume approach (Fig. 9).

$$\frac{dp}{dt} = -K\frac{vA - Q_{ri} - Q_s}{V_0 - sA} \tag{2}$$

where K is the fluid bulk modulus, v piston velocity, A piston cross sectional area, s piston displacement, V_0 piston volume at ODC, Q_{ri} the sum of the flow rates between of the ith displacement chamber and the pump ports and Q_S the sum of leakage through three different lubricating gaps between piston-cylinder (Q_{SK}), cylinder block-valve plate (Q_{SB}) and slipper-swash plate (Q_{SG}) (Ivantysyn and Ivantysynova, 2001).

The effect of change in relief groove geometry on the leakages through the lubricating gaps is negligible. As the optimization only changes the relief groove, the leakage in the lubricating gaps is assumed to remain the same for all the designs compared in the optimization and hence Q_S is eliminated from Eq. 2. Q_{ri} can be expressed as,

$$Q_{ri} = Q_{rHPi} + Q_{rLPi} \tag{3}$$

where Q_{rHPi} is the flow rate between the ith displacement chamber and the discharge port and Q_{rLPi} the suction port and can be expressed using the orifice flow relationship.

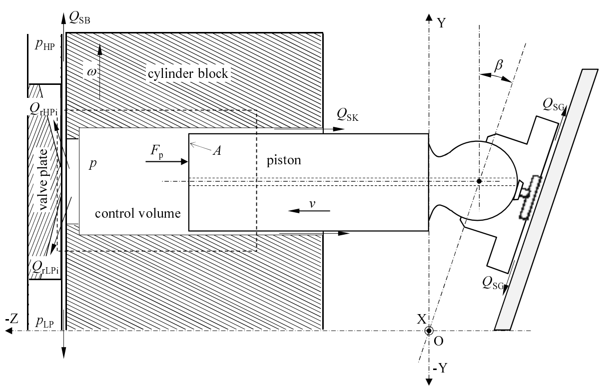

Fig. 9: *Control volume for evaluating the pressure inside the displacement chamber*

$$Q_{rHPi} = \alpha_D A_{rHP}\sqrt{\frac{2}{\rho}\left|p_{HP} - p_i\right|}\,\mathrm{sgn}\left(p_{HP} - p_i\right) \tag{4}$$

$$Q_{rLPi} = \alpha_D A_{rLP}\sqrt{\frac{2}{\rho}\left|p_{LP} - p_i\right|}\,\mathrm{sgn}\left(p_{LP} - p_i\right) \tag{5}$$

where α_D is the discharge coefficient, A_{rHP} and A_{rLP} is the valve plate opening areas available for flow transfer between the displacement chamber and the pump ports (Fig. 5), ρ fluid density, p_{HP} pressure at pump outlet and p_{LP} pressure at pump inlet.

V_0 in Eq. 2, the volume of the displacement chamber at the ODC when the piston is fully drawn out of the cylinder block, can be expressed as

$$V_0 = V_D + AR\left(\tan\beta_{max} + \tan\beta\right) \tag{6}$$

where V_D is the dead volume in the displacement chamber and β_{max} the maximum swash plate angle. The flow rate at the pump outlet (Q_{HP}) and inlet (Q_{LP}) are obtained by summing up the flow rate from the individual displacement chambers.

$$Q_{HP} = \sum_{i=1}^{z} Q_{rHPi} \tag{7}$$

$$Q_{LP} = \sum_{i=1}^{z} Q_{rLPi} \tag{8}$$

The pressure in the displacement chamber is further used to evaluate the forces exerted on the swash plate and the moments created due to the forces. According to Ivantysyn and Ivantysynova (2001), the total force on each piston can be expressed as,

$$F_{Ai} = F_p + F_f + F_2 \tag{9}$$

where F_{Ai} is the sum of the pressure force (F_p), friction force between the piston and the cylinder block (F_f) and the force due to acceleration of the piston (F_a). The normal components of the force due to each piston on the swash plate F_{NSi} and F_{Nyi} are responsible for creating the moment about the X-axis (Fig. 10).

$$F_{NSi} = \frac{F_{Ai}}{\cos\beta} \tag{10}$$

$$F_{Syi} = -F_{NSi}\sin\beta = -F_{Ai}\tan\beta \tag{11}$$

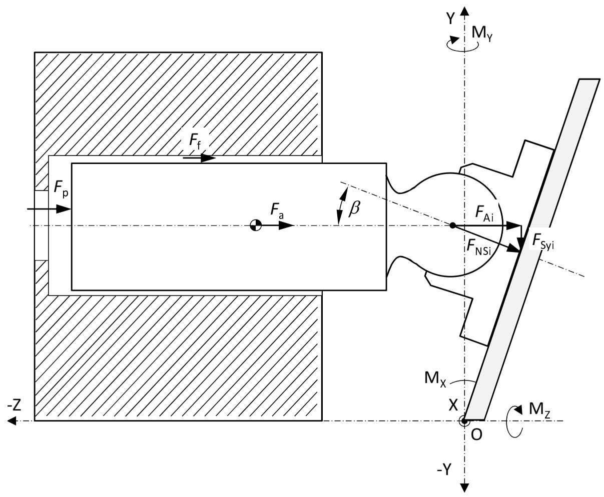

Fig. 10: *Moments on the swash plate*

The individual forces due to each piston can be summed up and the moment created due to these the forces about the coordinate axis can be evaluated as described below. The relations, in Eq. 12 to 14, determine the moments about the three coordinate axes and is dependent upon the pressure inside the displacement chamber (p), the angular position of each piston (φ_i) and the swash plate angle.

$$M_X = \frac{R}{\cos^2\beta}\sum_{i=1}^{z} F_{Ai}\cos\varphi_i \tag{12}$$

$$M_Y = R\sum_{i=1}^{z} F_{Ai}\sin\varphi_i \tag{13}$$

$$M_Z = -R\tan\beta\sum_{i=1}^{z} F_{Ai}\cos\varphi_i \tag{14}$$

The fluid density and bulk modulus used in this simulation are temperature and pressure dependent as expressed through the following relationships where bulk modulus K is defined as the reciprocal of the isothermal coefficient of compressibility (β_p).

$$\rho(T) = r_s\left(1 - a_{ls}T\right) \tag{15}$$

$$\rho(p,T) = \frac{\rho(T)}{1 - a_1\ln\left(\dfrac{a_2 + a_3T + p}{a_2 + a_3T}\right)} \tag{16}$$

$$K = \frac{1}{\beta_p(p,T)} = \frac{\rho(T)\left(a_2 + a_3T + p\right)}{a_3\rho(p,T)} \tag{17}$$

The values of the coefficients in Eq. 15 to 17 are listed here for oil type HLP32.

$a_1 = 0.07329654$

$a_2 = 1965.018$ bar

$a_3 = -2.968126$ bar/K

$r_s = 1047.03$ kg/m^3

$a_{ls} = 0.0005761668$ 1/K

The mathematical model is implemented using C++ programming language and differential equations are numerically solved using explicit Runge-Kutta method of order 5 due to Dormand & Prince with step size control. The C++ class for the implementation of Runge-Kutta method used in this work is taken from Ashby (2002).

The steps followed in the optimization procedure are summarized below.

1 Create a solution space of initial designs. See Table 2 for the initial range chosen for each of the parameters for a specific case study.
2 Simulate all initial design at the highest operating speed, full displacement and lowest operating pressure. This operating condition has the highest possibilities of occurrence of cavitation and over-pressurization. This operating condition is denoted as ‘1’ in Fig. 11. All the eight corners represent the extreme operating conditions at which the pump/motor will be operating.

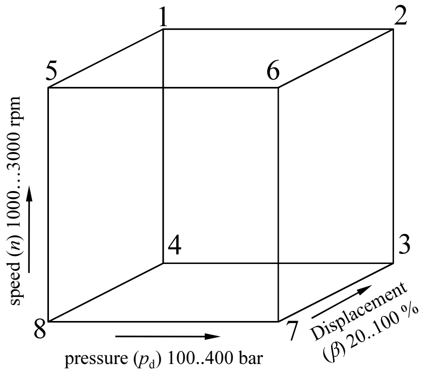

Fig. 11: *A cube enclosing the entire operating space in which the pump is expected to operate*

3 Fitness test all the initial designs against cavitation and over-pressurization. Remove the designs that fail the test from the solution space. This reduces the size of the solution space and hence the computation effort in the next steps.
4 Simulate remaining designs in the solution space at the rest of the operating corners. Designs are not evaluated at intermediate operating conditions on the assumption that the amplitudes of flow ripple and oscillating moments increase with increasing speed, pressure and displacement though not linearly. The assumption was made because of the authors’ prior knowledge about the sensitivity of the objectives to operating conditions.
5 At this stage, we have all the designs simulated at eight operating conditions and the objective can be evaluated. From each operating condition, a set of Pareto optimal/non-dominated designs are chosen. Deb (1999) describes the problem of multi-objective optimization in a clear and detailed manner. In a multi-objective optimization, a design is said to be non-dominated if (1) it is no worse than other designs in the solution space in all objectives and (2) better than other designs in at least one objective. In a multi-objective optimization with conflicting objectives it is hard to find a single optimal solution. The goal is to find multiple solutions that are Pareto optimal and then involve a higher-level decision making in selecting one among them as final optimal design (Deb 1999). This is the approach that is followed in this work in selecting the optimal solution. Second level non-dominated solutions were also kept along with first level Pareto optimal designs in the search for the optimal to keep diversity in parameter values (Srinivas and Deb, 1994). Pareto optimality was decided on the basis of three objectives ΔQ_{HP}, ΔQ_{LP} and ΔM_X. Q_{LK} was involved as a trade-off objective in the higher-level decision making during the final stages of selecting the optimal design. The higher level decision making is usually system specific and depends on preference of the designer for a certain design over the rest in final set. Figures 10 and 11 show a set of Pareto optimal designs selected and plotted against three objectives ΔQ_{HP}, ΔQ_{LP} and ΔM_X for a particular operating condition. All the designs are on the Pareto surface. But it is hard to visualize the

3D surface in the 2D plots. The dashed lines in Fig. 10 and 11 indicate the 2D Pareto front when considering only two objectives at a time, ΔQ_{HP} vs. ΔQ_{LP} or ΔQ_{HP} vs. ΔM_X. During the course of this research, it was observed that the profiles of M_Y and M_Z are strongly correlated to the profiles of flow ripples at high and low pressure ports. Consecutively, ΔM_Y and ΔM_Z have similar trend as ΔQ_{HP} and ΔQ_{LP}. It implies that designs with lower ΔQ_{HP} and ΔQ_{LP} would also have lower ΔM_Y and ΔM_Z. Hence ΔM_Y and ΔM_Z are not included explicitly in selecting the Pareto optimal designs.

6 From the sets of eight Pareto optimal designs, designs which are Pareto optimal in more than 2 operating conditions are grouped. This is done because the goal of the optimization is to find design(s) which would be quieter not just at one operating condition but at most of the operating range of the pump. It is mentioned as most of the operating range because a passive reduction technique like the precompression groove is sensitive to operating condition and is not expected to be optimal in all objectives at all operating conditions. The range of each parameter values are analyzed by the decision maker (DM). In an industrial setting, the pump designer would be the DM. The experience of the DM and DM's knowledge of the physical system is crucial in analyzing the designs. The analysis of the parameter values help in reducing the range of each parameter value closer to the optimal values. It also helps to reduce the solution space in the next cycle of optimization. In the first cycle of optimization, the optimal values of the parameters being unknown, the range of each parameter value is kept larger (refer Table 2). The first cycle helps largely in trying to identify a range for each parameter value which is closer to the optimum. In successive cycles, the range of each parameter value is much narrower and also closer to the optimal values. In the case study mentioned later in this work, four such cycles were involved before choosing a final design.

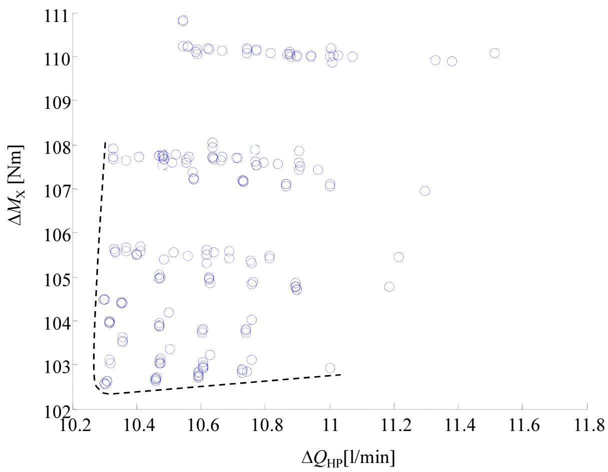

Fig. 12: *Plot of non-dominated designs with ΔQ_{HP} vs. ΔM_X*

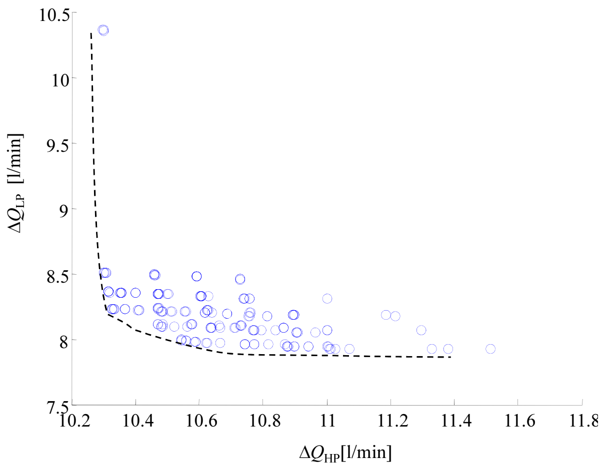

Fig. 13: *Plot of non-dominated designs with ΔQ_{HP} vs. ΔQ_{LP}*

7 The cycle of arriving at Pareto optimal designs and rearranging the range of parameter values is repeated until the DM feels any change in the range or interval of the parameter does not bring any improvement. The knowledge acquired by the DM at each generation helps in determining whether a satisfactory level of refinement of the parameters is reached or not. At the last cycle the DM has a final set of Pareto optimal designs which are optimal in more than 2 operating points. The task here for the DM is to choose a single design from the final set. Designs that remain after a satisfactory parameter refining are all Pareto optimal and can be termed as 'final set'. The MOGA methodology has an advantage that the DM is a part of the optimization procedure and continuously learns about the trend of objectives during each cycle. A disadvantage of this methodology is that the DM is confronted with not one but a set of final solutions that are optimal. The selection of the final design needs to be done on the basis of how well final set of designs perform against each other in the entire operating range. Also the decision to pick a final design is based on trade off between the objectives that is application specific.

The main goal of this optimization procedure is to help the pump designer arrive at the final set of optimal designs. The pump designer has the freedom to choose one design from the final set depending on system specific requirements or trade-offs. Choosing a final design from a final set of optimal design depends on specific system and the application where the pump/motor will be used. The process of choosing a final design from a final set for an industrial pump is detailed in Seeniraj (2009).

4 Case Study

An axial piston pump of size 75 cc having 9 pistons was chosen for the case study. The pump operating range was defined to be between pressure differential of 100 and 400 bar, rotational speed 1000 and 3000 rpm and swash plate displacement 20 and 100 %. In this case study, the objective function was formulated with four objectives namely ΔQ_{HP}, ΔQ_{LP}, ΔQ_{LK} and ΔM_X neglecting ΔM_Y and ΔM_Z for reason mentioned in section 3. A set of optimal precompression groove parameters was selected by subjecting the initial valve plate through Multi-objective Genetic Algorithm. Table 2 lists the initial parameter ranges with which the optimization was started and the final optimized values.

Table 2: List of precompression groove parameters

Relief groove parameters	Initial parameter range (x^l-x^h)	Optimal parameter value	Unit
Starting location of groove 1 (φ_{s1})	351-360	351.5	[°]
Ending location of groove 1 (φ_{e1})	4-8	7.5	[°]
Starting location of groove 2 (φ_{s2})	180-188	187.5	[°]
Ending location of groove 2 (φ_{e2})	171-177	176	[°]
Starting location of groove 3 (φ_{s3})	173-182	173	[°]
Ending location of groove 3 (φ_{e3})	184-188	185	[°]
Starting location of groove 4 (φ_{s4})	0-8	7.5	[°]
Ending location of groove 4 (φ_{e4})	355-360	355	[°]
Slope of groove 1 (m_1)	0.0-0.3	0.25	[mm^2/°]
Slope of groove 2 (m_2)	0.0-0.3	0.1	[mm^2/°]
Slope of groove 3 (m_3)	0.0-0.3	0.2	[mm^2/°]
Slope of groove 4 (m_4)	0.0-0.3	0.1	[mm^2/°]

A prototype valve plate was manufactured from the final optimal parameters. The prototype valve plate was compared in terms of sound pressure levels with an industrial valve plate design. The industrial design chosen for comparison represents a solution which was obtained by the manufacturer using an expensive trial and error approach involving the use of a simplified computational model and large amounts of noise measurements over a period of several years. The chosen industrial design is one of the quietest pumps of the selected type of swash plate machine.

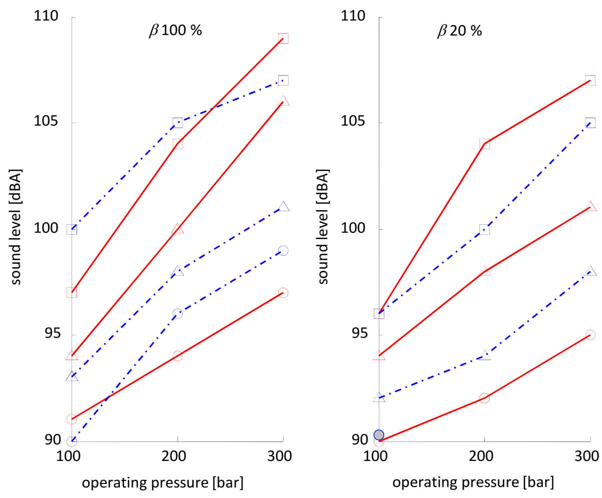

Fig. 14: *Sound level in dB for the industrial design (solid) and computer optimized (dash-dot) at 1000 rpm (circle), 2000 rpm (triangle) and 3000 rpm (square); at three operating pressures and two displacement (100 % and 20 %)*

The sound pressure level comparisons (Fig. 14) show that the computer optimized design was quieter than the industrial design in most of the operating conditions proving the effectiveness of the optimization procedure. It was interesting to note that in the case study, the final set of optimal designs were close to the industrial design in terms of the parameter values. This also confirms the effectiveness of the optimization procedure.

5 Conclusions

A simulation based optimization procedure has been proposed and tested for simultaneous optimization of both sources of noise (SBNS and FBNS) in an axial piston pump. Six objectives are considered in the optimization procedure which represent the noise sources (FBNS and SBNS) and volumetric efficiency. The implementation of the optimization procedure is explained with the help of a parameterized valve plate. The parameterization of the precompression groove has been explained. An important contribution of the parameterized approach is the optimality conditions on compression and expansion, explained in section 2, are simultaneously optimized unlike most previous works which consider compression and expansion separately. The proposed optimization procedure has been validated successfully with a case study and noise level measurements performed using an industrial pump.

Nomenclature

α_D	Orifice discharge coefficient	[-]
β	Pump displacement	[%]
Δ	Peak-to-peak variation	[-]
φ	pump rotation angle	[°]
ρ	Fluid density	[kg/m^3]
A	Piston cross sectional area	[m^2]
A_{rHP}	Flow area available between displacement chamber and discharge port	[m^2]
A_{rLP}	Flow area available between displacement chamber and suction port	[m^2]
F_{pi}	Pressure force exerted by single piston on the swash plate	[N]
K	Fluid bulk modulus	[Pa]
M_X, M_Y, M_Z	Swash plate moment about the respective axes	[Nm]
p	Displacement chamber pressure	[bar]
p_{HP}	Discharge port pressure	[bar]
p_{LP}	Suction port pressure	[bar]
Q_{HP}	Pump discharge flow rate	[m^3/s]
Q_{LK}	Pump case flow rate	[m^3/s]
Q_{LP}	Pump suction flow rate	[m^3/s]
Q_{rHPi}	Piston discharge flow rate	[m^3/s]
Q_{rLPi}	Piston suction flow rate	[m^3/s]
R	Piston pitch radius	[m]
V_0	Piston volume at ODC	[m^3/s]
V_D	Piston dead volume	[m^3/s]
x_i^h	Upper limit for parameter i	[-]
x_i^l	Lower limit for parameter i	[-]
z	Number of pistons	[-]

Acronym

DM	Decision maker
FBNS	Fluid borne noise source
IDC	Inner dead center
MOGA	Multi objective genetic algorithm
ODC	Outer dead center
PCFV	Precompression filter volume
SBNS	Structure borne noise source

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Ganesh Kumar Seeniraj
Born on May 10, 1980 in Sivakasi, Tamil Nadu, India. He received his Bachelor of Engineering in Mechanical Engineering from College of Engineering, Guindy, Anna University, India in 2001. He received his MS in Mechanical Engineering from Kettering University, USA in 2003 and PhD from Purdue University, USA in 2009. Improving the efficiency of Fluid Power Systems at both component and system level has been his main research area. His research focus also includes reduction of fluid borne and structure borne noise sources in hydraulic pumps/motors and transmissions.

Monika Ivantysynova
Born on December 11th 1955 in Polenz (Germany). She received her MSc. Degree in Mechanical Engineering and her PhD. Degree in Fluid Power from the Slovak Technical University of Bratislava, Czechoslovakia. After 7 years in fluid power industry, she returned to university. In April 1996 she received a Professorship in fluid power & control at the University of Duisburg (Germany). From 1999 until August 2004 she was Professor of Mechatronic Systems at the Technical University of Hamburg-Harburg. Since August 2004 she is Professor in Mechanical Engineering and Agricultural and Biological Engineering at Purdue University, USA. She was approved as Maha named Professor in Fluid Power Systems and director of the Maha Fluid Power Research Center at Purdue University in November 2004. Her main research areas are energy saving actuator technology and model based optimization of displacement machines as well as modeling, simulation and testing of fluid power systems. Besides the book "Hydrostatic Pumps and Motors" published in German and English, she has published more than 90 papers in technical journals and at international conferences.