Noise reduction of hydraulic systems by axial piston pumps with variable reversing valves

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Abstract

Hydraulic pumps are often considered to be the leading noise source. The main criteria for noise generation of hydraulic piston pumps are the flow ripple, the pulsating piston force and the pulsating swash plate torque.

Common methods to reduce noise of axial piston pumps are grooves or boreholes to smoothen the reversing process. More recently, pre-compression volumes are used which mainly focus on flow ripple reduction. However, these methods only have the ability to reduce noise and flow ripple significantly in a designed operating range.

Using special reversing valves, a variable approach to reduce noise is investigated in this paper. Hereby, the valve openings can be adapted to the actual operating point of the pump, which is defined by the rotational speed, the pump displacement and the system pressure. Furthermore, this variability allows different control strategies, so that the focus can be shifted between the different noise criteria and optimized for a specific hydraulic system. The required valve openings for different control strategies and operating points were determined by simulation runs and verified by measurements.

Depending on the investigated operating point, flow ripple reductions of up to 50% and swash plate torque ripple reductions of up to 70% were measured. Furthermore, different cylinder pressurization slopes are presented along with the resulting pump noise. Depending on the operating point, sound power level of the pump itself can be reduced by up to 2dBA compared to a highly optimized standard pump.

KEYWORDS: flow ripple, noise, axial force, swash plate torque, axial piston pump

1. Noise creation in hydraulic systems

The creation of air-borne noise in a hydraulic system is shown in Figure 1.



Figure 1: Noise creation in a hydraulic system /Mue02/

In hydraulic systems the driving engine, the hydraulic pump and the valves are sources of noise. The induced excitations are partially emitted directly as air-borne noise whereas another part is transmitted into the hydraulic system as structure-borne noise as well as fluid-borne noise. This way other parts of the hydraulic system are forced to vibrate and themselves emit air-borne noise. Often the hydraulic pump is considered to be the main noise source in hydraulic systems /Goe08/.

In general, methods for noise reduction can be divided into primary and secondary methods. Secondary methods reduce noise after its creation and are generally located outside the pump. Primary methods avoid the creation of noise and are typically located inside the hydraulic pump. The configuration of the valve plate plays a key role in primary noise reduction, but also affects other important pump characteristics as shown in **Figure 2**.





Influencing the flow ripple, the valve plate affects the pulsation of the delivery pressure, leading to vibrations in the hydraulic circuit. Influencing the cylinder pressure profile, the oscillation of the axial force and of the swash plate torque is affected. These oscillations force the pump to emit air-borne noise /Mue02/, /Joh05/. Furthermore, oscillation of the swash plate torque generates control pressure ripples, affecting controllability of pumps with variable displacement. This paper focuses on flow ripple, axial force and swash plate torque. Efficiency and cavitation are considered as restraints.

In order to determine a suitable variable reversing system, a theoretical approach is presented by separately investigating each of the three criteria for noise creation. Hereby two different valve plate configurations where investigated analytically as well as in simulation runs. Furthermore, these systems where built and run on a specially designed test bench. Beneath the typical measurements, flow ripple was measured using an anechoic ending and cylinder pressure was measured using a telemetric system inside the pump housing /Naf11/. In the following, a promising system in terms of practicability is presented: A valve plate with variable reversing valves.

2. The variable reversing valves

The reversing valves allow for a continuously variable flow area between the cylinder kidney and the delivery or suction port. **Figure 3** shows the principle of the variable valve plate configuration and the possible variation of the valve opening as a function of the rotation angle.



Figure 3: Principle of the variable reversing valves

The maximum valve opening can be varied, whereas the gradient $\partial A_v/\partial \phi$ is fixed by the borehole diameter. The valve openings are defined as positive when they connect the cylinder to the upcoming kidney. They are defined as negative when the cylinder is connected to the previous kidney. Generally a positive opening accelerates the

pressure change and a negative opening delays the pressure change. Due to the overlap between delivery and suction kidney, a negative opening decreases efficiency.



The concrete design of the reversing valves is shown in Figure 4.

Figure 4: Design of the variable reversing valves

Depending on the position of the valve piston, the reversing area is connected to the suction or the discharge port. This position is varied by a torque motor that drives a mechanical stop. A retaining spring ensures a secure piston position. This design was chosen to reduce the sensitivity of the valve piston position to disturbing forces such as transient and static flow forces occurring during the reversing processes.

These reversing valves are fast enough to react on a change in operating conditions. Best adaption can be provided when a change in the set value is known beforehand as the adjustment can be made simultaneously to the change in operating point.

3. Functionality of the reversing valves

In order to investigate such an actively controlled valve plate configuration, a simulation model in AMESim was built and verified with cylinder pressure and flow ripple measurements /Naf11/. Therefore a specific test stand with an anechoic ending and a pressure transducer in the piston barrel was built.

Figure 5 shows the influence of different valve opening areas at the ODC and the IDC on the shape of the cylinder pressure profile as well as sound power level.



Figure 5: Functionality provided by the reversing valves (β '=100%, n=1500rpm, p_{HD}=200bar)

In both dead centre areas, a positive valve opening results in an earlier pressure change whereas a negative opening delays the pressure change. The changes in the sound power level are referenced on the value $L_{WA,Ref}$ which occurs at $A_{ODC}/A_{max} = 0.3$ and $A_{IDC}/A_{max} = 0.1$. These valve settings with an early start of the pressure change at low pressure gradient result in a low sound pressure level. A late and sharp pressurization ($A_{ODC}/A_{max} = -0.5$) significantly increases the sound power level by 5.4dBA. At an early pressurization ($A_{ODC}/A_{max} = 0.7$) the delivery pressure is reached too early resulting in a pressure peak. Sound power level however only rises by 0.6dBA. An early but sharp depressurization ($A_{IDC}/A_{max} = 0.5$) increases sound power level by 3.4dBA while a late and sharp depressurization ($A_{IDC}/A_{max} = -0.7$) increases this value by 8.7dBA. These enormous changes in sound power level emphasise the necessity of a very careful layout of pressure relief grooves and in this case the valve areas.

Due to the variable reversing process at the ODC as well as at the IDC, all kinds of different control strategies become possible. The valve settings for the different strategies were found by varying both valve openings independently in automated simulation runs as shown in **Figure 6**.



Figure 6: Structure of automated simulation runs

This way a matrix was derived for each investigated operating point containing the design criterion as function of the different combinations of the valve settings. The settings resulting in the best criterion value where chosen for the further investigations.

In this paper two strategies are presented. The "swash plate torque strategy" concentrates on the noise being directly emitted by the pump, whereas the "flow ripple strategy" focuses on flow ripple reduction which results in noise being emitted by the hydraulic system.

3.1. Swash Plate Torque Strategy

As outlined above, the direct pump noise is mainly influenced by the oscillation of the axial piston force and the swash plate torque. Ideally, a sinusoidal shape of the axial force profile is desired whereas for a minimization of the swash plate torque oscillation requires a triangular shape of the axial force profile. Simulative investigations have shown that neither of them can be perfectly achieved in realistic valve plate configurations /Naf11/. Therefore the peak-to-peak swash plate torque is used as design criteria in this control strategy. This way an axial force profile with lower excitation frequencies is implied. The required valve openings were determined as described above. **Figure 7** shows the resulting peak-to-peak swash plate torque amplitudes compared to the standard valve plate configuration at a constant speed of 1500 rpm.



Figure 7: Measurement results of the pump noise strategy and the standard pump at constant speed (n=1500rpm)

These swash plate torque values were derived from measured cylinder pressure values. The swash plate torque ripple of the standard pump increases with the delivery pressure. It slightly increases with rising pump displacement whereas the gradient $\partial(\Delta M_{SW})/\partial p_{HD}$ remains almost unaffected. This gradient significantly decreases with decreasing pump displacement. Therefore, best improvements are achieved at higher discharge pressures and lower pump displacements. This way the swash plate torque of the standard pump can be lowered by more than 70% at $\beta' = 25\%$, $p_{HD} = 300$ bar and n = 1500rpm. This effect is shown more detailed in **Figure 8** showing a comparison of the measured oscillation of the swash plate torque as a function of the rotation angle at different pump displacements.



Figure 8: Swash plate torque at different displacements (n=1500 rpm, p_{HD}=300 bar)

Beneath the swash plate torque ripple, the sound power level is affected by this control strategy. In **Figure 9** the changes in sound power level ($L_{WA;VVU} - L_{WA;std}$) compared to the standard configuration are shown for different operating conditions.



Figure 9: Change in sound power level compared to standard configuration

At constant speed (1500rpm) the sound power level could be mainly reduced at medium pressure levels. At constant pump displacement (100%) the sound power level was reduced at almost all pressure and speed variations. At medium pressures, improvements up to 2dBA were achieved. Due to the optimization of the standard grooves for industrial speeds (1500rpm) and high pressures, the sound power level of the VVU is higher than that of the standard configuration in this operating range. This is due to the general advantage of grooves compared to simple holes. The VVU could however be further improved by introducing little grooves preceding the holes.

At low pressure, a reduction of the swash plate torque ripple comes along with a reduction of the sound power level. With rising delivery pressure, the swash plate torque can be further reduced. However, the improvement in sound power level is reduced. This indicates that the influence of swash plate torque ripple on the sound power level is reduced for increased pressures. This might be due to the low hydraulic cushion of the swash plate bearing of the used test pump compared to other hydraulic pumps of the swash plate type. This way, more damping in the swash plate bearing occurs. At pumps with a higher hydraulic cushion a higher influence on the pump noise is expected.

3.2. Flow Ripple Strategy

As described above, noise emission in hydraulic systems is strongly affected by fluid borne noise emitted by the hydraulic pump. Therefore, the following control strategy aims to reduce the pump flow ripple. The settings of the flow ripple strategy were derived in automated simulation runs for each investigated operating point as described above. The resulting measurements are shown in **Figure 10** in comparison to the standard valve plate configuration for a constant speed of 1500rpm.



Figure 10: Measurements of the flow ripple strategy and the standard pump (n=1500rpm)

The pressure ripple of the standard pump increases with the delivery pressure and the pump displacement. This characteristic does not basically change for the flow ripple strategy using the reversing valves, whereas the gradient $\partial(\Delta p_{HD})/\partial p_{HD}$ is generally smaller and decreases with rising delivery pressure. Therefore, the pressure ripples are especially reduced at higher delivery pressures compared to the standard configuration. For discharge pressures below "piston pressure", the standard pressure ripples can not be improved. However, in most operating conditions, the reversing valves outperform the standard configuration. At lower pump speeds, improvements get even better and pressure ripple reductions of more than 50% are possible.

Figure 11 shows the measured pressure ripple as a function of the rotation angle at different pump displacements in a more detailed manner.



Figure 11: Pressure ripple at different pump displacements (n=1500rpm, p_{HD}=300bar)

4. Conclusion

Noise emission caused by hydraulic piston pumps is mainly influenced by the flow ripple as well as the oscillation of the axial force and the swash plate torque. These three criteria are strongly influenced by the reversing process and therefore by the valve plate configuration. In order to achieve the best performance in all operating conditions, a variable valve plate configuration is desired. Therefore a configuration with variable reversing valves in the IDC and ODC was designed. Due to the active adjustability of the valve openings, different control strategies are possible. In this investigation, the swash plate torque strategy and the flow ripple strategy were described in simulation and hardware tests.

Measurements of the flow ripple strategy show a significant improvement of the delivery pressure ripple in comparison to the standard configuration over a wide range of operating points. Applying the swash plate torque strategy, the oscillation of the swash plate torque can be significantly reduced. Especially at high pressures, low pump displacements and lower speeds, significant improvements were achieved. In

particular, the flow ripple was reduced by up to 50% and the peak-to-peak swash plate torque was reduced by up to 70%. The sound power level of the pump was reduced by up to 2dBA.

In future, the presented strategies could be implemented in a control unit, adjusting the valve openings depending on the actual operating conditions. Furthermore, intelligent control algorithms are desirable in order to optimize noise emission at noise sensitive places such as a driver's cab in a mobile machine or an operator station of any other hydraulic system.

5. References

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6. Nomenclature

β	pump displacement angle	0
β'	pump displacement	%
φ	rotation angle of the cylinder	o
$oldsymbol{arphi}_{ extsf{HD}}$	start angle of the delivery kidney	0
$A_{ ho}$	cross sectional piston area	M²
A_V	valve area	M²
f _p	piston frequency	1/s
L _{wA}	sound power level	dBA
M _{sw}	swash plate torque	Nm
p _{cyl}	cylinder pressure	bar
р нд	delivery pressure	bar
p_{p}	"piston pressure"	bar
R	radius of the piston pitch circle	m
Vo	cylinder volume at zero displacement	m³
Δx	peak-to-peak value of x	
X	oscillation of x around its mean value	
IDC	inner dead centre	
ODC	outer dead centre	