On Fluid Power Pump and Motor Design

Tools for Noise Reduction
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Liselott Ericson

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To my parents

Marita & Hans

Never get so busy making a living,
that you forget to make a life

Anonymous
Abstract

Noise and vibration are two of the main drawbacks with fluid power systems. The increasing requirements concerning working environment as well as machines’ impact on surroundings put components and systems to harder tests. The surrounding machines, e.g. combustion engines, have made considerable progress regarding the radiated noise. This allows the fluid power system’s noise to become more prominent. Noise from fluid power systems has been a research topic for several decades and much improvement has been achieved. However, considerable potential for improvement still remains.

In addition to the legislation governing working environment, the machines tend to be used as more multi-quadrant machines, which require more flexible noise reduction features. One of the main benefits with fluid power is the high power density. To increase this value even more, the system’s working pressure increases, which correlates with increased noise level.

The main source of noise is considered to be the pump and motor unit in the fluid power system. The noise can be divided into two parts: fluid-borne noise and structure-borne noise. The fluid borne noise derives from flow pulsation which is subsequently spread through pipeline systems to other parts of the fluid power systems. The flow pulsation is created due to the finite stiffness of oil and the limited number of pumping elements. The structure-borne noise generates directly from pulsating forces in the machine. The pulsating forces are mainly created by the pressure differences between high and low pressure ports.

Effective and accurate tools are needed when designing a quiet pump/motor unit. In this thesis simulation based optimisation is used with different objective functions including flow pulsation and pulsating forces as well as audible noise. The audible noise is predicted from transfer functions derived from measurements. Two kinds of noise reduction approaches are investigated; cross-angle in multi-quadrant machines and non-uniform placement of pistons. The simulation model used is experimentaly validated by source flow measurements. Also, source flow measurements with the source admittance method are investigated.

In addition, non-linear flow through a valve plate restrictor is investigated and the steady state restrictor equation is proposed to be extended by internal mass term.
Acknowledgements

The project was conducted at the Division of Fluid and Mechatronic Systems\(^1\) at Linköpings University. Several people have been involved in this thesis: some have been more involved than others but all had to be there to make the project possible and enjoyable. My supervisor, Jan-Ove Palmberg, former Head of Division, I want to thank his time, his passion for the subject and our discussions. Special thanks go to Petter Krus, current Head of Division and to my co-author Johan Ölvander for his support. Here it is fitting to thank the rest of my colleagues, both past and present, at the Division of Fluid and Mechatronic Systems and Machine Design who have made my workdays enjoyable.

Thanks also go to Parker Hannifin AB for their financial support and also for the kind welcome during the days I spent in Trollhättan and also for the hardware support during the project. Special thanks to Andreas Johansson, who has always been interested in my work and given encouraging discussions.

The important parts of making my life enjoyable and secure I have saved to the end. First I would like to thanks my parents who have stood by me through thick and thin. Ludde, my loving dog, we have had many long and thoughtful strolls; probably we were not thinking about the same things during our walks but they have helped me to free my mind. Last but by no means least, Martin, who came and turned my life up side down but I would not want to change a second. You have gilded my life.

Linköping, December, 2011

Liselott Ericson

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\(^1\)Prior to 2010, the name of the division was Division of Fluid and Mechanical Engineering Systems
Papers

The dissertation is built on the work of following papers and will be referred to by their Roman numerals. Paper [I] investigates the possibility to use a cross-angle for machines working both as pump and motor with fixed pressure port. Paper [II] is about optimisation and in particular choosing the objective function when designing a quiet positive displacement machine. The structure-borne noise from the pump and motor housing is predicted and optimised in paper [III]. The method is further used when a non-uniform placement of pistons is optimised in paper [IV]. In paper [V], air bubbles in oil are considered when different operating conditions and valve plates are used. This paper was selected at 7th JFPS International Symposium on Fluid Power for publication in the JFPS International Journal of Fluid Power Systems. Paper [VI] shows how the orifice’s steady state equation at the valve plate port openings can be modified to make the equation more suitable for the dynamic flow which is produced in hydraulic motors and pumps. The main author is the first author in all the papers, supervised by the co-authors.

The papers have been corrected for printing errors and the layout of text and figures has been changed for uniform appearance throughout the thesis. A short summary of each paper can be found in chapter 10.

Papers [VII], [VIII], [IX], and [XI] are not included but still important part of the work. The first two papers mentioned are about a flow pulsation measurement method; the source admittance method. Paper [XI] is about an electric hybrid vehicle with a hydraulic energy recovery system. This paper is written by the first-mentioned author while the next two authors were mainly involved in building the vehicle and supervised by the fourth author. Publication [X] is a licentiate thesis published in 2008.

Appended papers


Other papers and publications


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# Nomenclature

The nomenclature shows the majority of the parameters and variables used in the thesis. The equation or page number shows the parameter’s first appearance.

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<td>Restrictor area</td>
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<td>[m$^2$]</td>
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<td>A-weighting filter</td>
<td>(4.11)</td>
<td>[dB]</td>
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<td>$B$</td>
<td>Four-pole element</td>
<td>(5.3)</td>
<td>[m$^3$]/Ns</td>
</tr>
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<td>$B_p$</td>
<td>Four-pole element</td>
<td>(5.19)</td>
<td>[m$^3$]/Ns</td>
</tr>
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<td>$C$</td>
<td>Four-pole element</td>
<td>(5.3)</td>
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<td>[Ns/m$^3$]</td>
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<td>$C_q$</td>
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<td>$D$</td>
<td>Displacement</td>
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<td>[m$^3$/rev]</td>
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<td>Four-pole element</td>
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<td>[-]</td>
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<td>$F_c$</td>
<td>Axial piston force</td>
<td>(2.9)</td>
<td>[N]</td>
</tr>
<tr>
<td>$F_z$</td>
<td>Total axial piston force</td>
<td>(2.12)</td>
<td>[N]</td>
</tr>
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<td>$F$</td>
<td>Multi-objective function</td>
<td>(4.1)</td>
<td></td>
</tr>
</tbody>
</table>
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\( G \) \hspace{1em} \text{Constraints, equation (4.3)}

\( G_{Fz} \) \hspace{1em} \text{Transfer function, equation (6.1) \hspace{1em} \left[1/m^2 \right]}\)

\( G_{Mx} \) \hspace{1em} \text{Transfer function, equation (6.1) \hspace{1em} \left[1/m^3 \right]}\)

\( G_{My} \) \hspace{1em} \text{Transfer function, equation (6.1) \hspace{1em} \left[1/m^3 \right]}\)

\( J_0 \) \hspace{1em} \text{Bessel function of order 0, equation (5.5) \hspace{1em} [-]}\)

\( J_2 \) \hspace{1em} \text{Bessel function of order 2, equation (5.5) \hspace{1em} [-]}\)

\( K \) \hspace{1em} \text{Pressure relief groove area constant, equation (2.6) \hspace{1em} \left[m^2 \right]}\)

\( L \) \hspace{1em} \text{Hydraulic inductance, equation (3.14) \hspace{1em} \left[Ns^2/m^5 \right]}\)

\( L \) \hspace{1em} \text{Pipe length, equation (5.5) \hspace{1em} \left[m \right]}\)

\( L_A \) \hspace{1em} \text{A-weighted sound pressure level, equation (4.11) \hspace{1em} \left[dB(A) \right]}\)

\( L_{pn} \) \hspace{1em} \text{Sound pressure level, equation (4.9) \hspace{1em} \left[dB \right]}\)

\( L_{p} \) \hspace{1em} \text{Sound pressure level, equation (4.10) \hspace{1em} \left[dB \right]}\)

\( M \) \hspace{1em} \text{Mean value, equation (6.4)}

\( M_x \) \hspace{1em} \text{Moment on the valve plate, equation (2.13) \hspace{1em} \left[Nm \right]}\)

\( M_y \) \hspace{1em} \text{Moment on the valve plate, equation (2.14) \hspace{1em} \left[Nm \right]}\)

\( M_{axis} \) \hspace{1em} \text{Torque, equation (6.3) \hspace{1em} \left[Nm \right]}\)

\( N \) \hspace{1em} \text{Number of summarised harmonics, equation (4.10) \hspace{1em} [-]}\)

\( N \) \hspace{1em} \text{Viscous friction factor, equation (5.4) \hspace{1em} [-]}\)

\( P_0 \) \hspace{1em} \text{Pressure at position 0, equation (5.17) \hspace{1em} \left[Pa \right]}\)

\( P_1 \) \hspace{1em} \text{Pressure at pump flange, equation (5.1) \hspace{1em} \left[Pa \right]}\)

\( P_2 \) \hspace{1em} \text{Pressure at position 2, equation (5.3) \hspace{1em} \left[Pa \right]}\)

\( P_3 \) \hspace{1em} \text{Pressure at position 3, equation (5.27) \hspace{1em} \left[Pa \right]}\)

\( P_i \) \hspace{1em} \text{Pressure at position i, equation (5.25) \hspace{1em} \left[Pa \right]}\)

\( P_j \) \hspace{1em} \text{Pressure at position j, equation (5.25) \hspace{1em} \left[Pa \right]}\)

\( Q_0 \) \hspace{1em} \text{Flow at position 0, equation (5.17) \hspace{1em} \left[m^3/s \right]}\)

\( Q_1 \) \hspace{1em} \text{First harmonic of the compressible flow, equation (2.8) \hspace{1em} \left[m^3/s \right]}\)

\( Q_1 \) \hspace{1em} \text{Flow at pump flange, equation (5.1) \hspace{1em} \left[m^3/s \right]}\)

\( Q_2 \) \hspace{1em} \text{Flow at position 2, equation (5.3) \hspace{1em} \left[m^3/s \right]}\)

\( Q_i \) \hspace{1em} \text{Flow at position i, equation (5.25) \hspace{1em} \left[m^3/s \right]}\)
Flow at position j, equation (5.25) \[ \text{m}^3/\text{s} \]

Kinematic flow, equation (2.3) \[ \text{m}^3/\text{s} \]

Theoretical pump flow, equation (2.8) \[ \text{m}^3/\text{s} \]

Source flow, equation (5.1) \[ \text{m}^3/\text{s} \]

Source flow at valve plate, equation (5.17) \[ \text{m}^3/\text{s} \]

Source flow at pump flange, equation (5.23) \[ \text{m}^3/\text{s} \]

Gas constant, equation (3.18) \[ \text{J/kgK} \]

Resistance, equation (3.14) [-]

Radius of barrel, equation (2.10) \[ \text{m} \]

Vapour condensation rate, equation (3.21) [-]

Vapour generation rate, equation (3.20) [-]

Radius of piston mounting at axis, equation (3.2) \[ \text{m} \]

Radius of piston, equation (3.2) \[ \text{m} \]

Linearised restrictor coefficient, equation (5.7) \[ \text{Ns/m}^5 \]

Parameter space, equation (4.1)

Absolute temperature, equation (3.18) \[ \text{K} \]

Wave propagation time, equation (5.4) \[ \text{s} \]

Period time, equation (2.7) \[ \text{s} \]

Cylinder dead volume, equation (3.2) \[ \text{m}^3 \]

Volume behind discharge channel, equation (5.18) \[ \text{m}^3 \]

Cylinder volume, equation (3.1) \[ \text{m}^3 \]

Transfer function, equation (5.20) [-]

Transfer function, equation (5.20) \[ \text{Ns/m}^5 \]

Fourier transformed objective, equation (4.7)

Impedance at valve plate, equation (5.17) \[ \text{Ns/m}^5 \]

Point impedance at pump flange, equation (5.2) \[ \text{Ns/m}^5 \]

Termination impedance, page (68) \[ \text{Ns/m}^5 \]

Characteristic pipe impedance, equation (5.4) \[ \text{Ns/m}^5 \]

Point impedance at position i, equation (5.25) \[ \text{Ns/m}^5 \]
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\[ Z_j \] Point impedance at position \( j \), equation (5.25) \( \text{[Ns/m}^5\text{]} \)

\[ Z_s \] Source impedance, equation (5.1) \( \text{[Ns/m}^5\text{]} \)

\[ Z_{s1} \] Source impedance at pump flange, equation (5.23) \( \text{[Ns/m}^5\text{]} \)

\[ a \] Speed of sound, equation (5.5) \( \text{[m/s]} \)

\[ d_h \] Hydraulic diameter, equation (3.8) \( \text{[m]} \)

\[ f \] Frequency, equation (5.16) \( \text{[rad]} \)

\[ f \] Single-objective function, equation (4.1)

\[ f_g \] Gas mass fraction, equation (3.20) \( \text{[-]} \)

\[ f_v \] Vapour mass fraction, equation (3.20) \( \text{[-]} \)

\[ g \] Constraint, equation (4.2)

\[ h \] Height of pressure relief groove, page (22) \( \text{[m]} \)

\[ h_{11} \] Admittance matrix element, equation (5.30) \( \text{[m}^3\text{/Ns]} \)

\[ h_{12} \] Admittance matrix element, equation (5.30) \( \text{[m}^3\text{/Ns]} \)

\[ h_{21} \] Admittance matrix element, equation (5.30) \( \text{[m}^3\text{/Ns]} \)

\[ h_{22} \] Admittance matrix element, equation (5.30) \( \text{[m}^3\text{/Ns]} \)

\[ i \] Imaginary unit, equation (5.5) \( \text{[-]} \)

\[ i \] Integer, equation (4.3) \( \text{[-]} \)

\[ j \] Integer, equation (2.1) \( \text{[-]} \)

\[ k \] Turbulent kinetic energy, equation (3.20) \( \text{[J]} \)

\[ k_c \] Linearised flow-pressure coefficient, equation (5.18) \( \text{[m}^3\text{/Ns]} \)

\[ k_{c1} \] Linearised flow-pressure coefficient, equation (5.7) \( \text{[m}^3\text{/Ns]} \)

\[ l \] Length between pressure transducers, equation (5.13) \( \text{[m]} \)

\[ l_e \] Effective length, equation (3.8) \( \text{[m]} \)

\[ l_r \] Restrictor length, equation (3.8) \( \text{[m]} \)

\[ m \] Integer, equation (2.3) \( \text{[-]} \)

\[ m \] Mass of free gas, equation (3.18) \( \text{[kg]} \)

\[ m_j \] Mass of the internal mass, equation (3.7) \( \text{[kg]} \)

\[ n \] Area exponent, equation (2.5) \( \text{[-]} \)

\[ n \] Integer, equation (4.2) \( \text{[-]} \)
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<th>Symbol</th>
<th>Definition</th>
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<td>(5.7)</td>
<td>[Pa]</td>
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<td>Pressure drop over mass</td>
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<td>[Pa]</td>
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<td>[m$^3$/s]</td>
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<td>$v$</td>
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x
Fz point of action, equation (2.15) [m]
x
Optimisation parameter, equation (4.1)
x0
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x;
Cylinder position, equation (2.10) [m]
y
Fz point of action, equation (2.16) [m]
y;
Cylinder position, equation (2.11) [m]
z
Number of pistons, equation (2.1) [-]
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Maximum displacement angle, equation (3.2) [rad]
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βc
Efficient bulk modulus, equation (2.5) [Pa]
βe,a
Effective bulk modulus, adiabatic process, equation (3.19) [Pa]
βe,i
Effective bulk modulus, isothermal process, equation (3.18) [Pa]
βoil
Efficient Bulk modulus of oil without air, equation (3.18) [Pa]
δc
Compressible non-uniformity, equation (2.5) [-]
δk
Kinematic non-uniformity, equation (2.4) [-]
ε
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ε
Displacement fraction, equation (2.1) [-]
φ
Angular cylinder position, equation (2.6) [rad]
γ
Cross-angle, equation (3.2) [rad]
γ
End correction’s factor, equation (3.8) [-]
η
Dead centre angle, equation (7.6) [rad]
κ
Polytropic exponent, equation (3.19) [-]
λ
Weighting value, objective functions, equation (4.3)
ϑ
Discrimination value, constraints, equation (4.3)
ρ
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ρl
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ρv
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1

Introduction

Fluid power systems are mainly used to transmit energy from one source to various kinds of functions. Fluid power is used in many mobile applications, such as passenger cars, cranes, wheel loaders, and excavators, and also in manufacturing industries. The fluid power systems’ main task is to convert energy from mechanical input energy to mechanical output energy via hydraulic energy, refers to transmissions. Usually, the hydraulic energy is created by a pump which receives mechanical energy from combustion engines in mobile applications or electric motors in industries. A motor or a cylinder is then used to convert the hydraulic energy back to useful mechanical energy depending on the function’s requirements.

There are several benefits and drawbacks with hydraulic systems. The main benefits are the power density, robustness, and the flexibility of the systems while the drawbacks are noise and vibration, which partially go hand in hand. These are generated due to the high output power and the demand to enhance the power compactness; operation pressure is increased which correlates to an increase in noise and vibrations. The demands on multi-quadrant machines are also increased when the hydraulic systems are used to regenerate energy. For mobile hydraulic systems, the problem has increased due to the reduced noise in combustion engines. Hydraulic systems have not improved to the same extent and the hydraulic noise is more apparent today than 20 years ago.

The machines, i.e. pumps, motors and transformers, are considered to be the main source of noise in hydraulic systems. The machines produce a superimposed flow ripple which is spread through the hydraulic pipeline system. The discontinuities interact with the flow ripple and produce pressure pulsations, which are further converted into structure vibrations and later into audible noise. Also, forces inside a machine produce vibration in the structure and hence audible noise.
1.1 Research areas and needs

The research on noise in hydraulic systems can be divided into different areas where the focus diverges slightly:

Inside pump/motor unit
Reduce noise where it is created, e.g. reduction of flow and force pulsations.

Pump/motor housing
Material and design consideration to reduce vibration and noise emission at the housing.

System design
System design consideration to reduce pressure pulsations and vibration in the system mainly caused by the pump/motor unit.

Simulation technology
Due to the high frequency and the complexity of the behaviour, the simulation technology have high demands. Also, optimisation methods of machines and systems are developed.

Measurement
Measurement technology is used for validation and to get better understanding and knowledge about the products.

Most researchers agree on one thing: The main source of noise is the motor/pump units in the hydraulic systems. Mainly the tool areas, simulation and measurement, are dealt with in this project. In addition, the tools are used to investigate different noise reduction measures inside the machines.

The need for quiet machines is increasing due to the increased competition from other related fields. Also, new operational fields have opened their eyes to fluid power technology and see its potential. In addition, the fluid power noise has become more prominent due to the reduction of other sources of noise, e.g. combustion engines. Work environment requirements are also increasing. For all these reasons, it is very important to reduce the noise produced in fluid power machines. Good, effective, and accurate tools are needed to make well elaborated investigations and decisions when the noise produced should be minimised.

1.2 Aims

This thesis aims at contributing to noise reduction in fluid power pumps and motors. As most noise in fluid power systems is judged to have its origin from pumps or motors the fluid power system as a whole will be quieter as well. The aim is to devise a general strategy to minimise as far as possible the
pump/motor unit’s noise impact on the rest of the system, whatever system the unit is mounted in.

1.3 Contribution

Effective, accurate and reliable tools are needed to be able to effectively design pump/motor units from a noise point of view. Simulation based optimisation is used throughout the thesis, which also shows how it can be used. Many different objectives are considered including audible noise prediction through transfer functions. Different kinds of ordinary noise reduction features are used to evaluate the optimisation strategy. The thesis contributes to a better understanding of how to formulate an objective function dependent on as well as independent of the knowledge of the system the unit should be used in.

The thesis treats multi-quadrant machines which means machines working as both pump and motor. The contribution includes better understanding of motor and pump operations as well as how to decrease noise in such machines.

Two noise reduction features are investigated in addition to the ordinary features. Cross-angle is investigated in multi-quadrant machine. Non-uniform placement of pistons are also investigated. The benefits and disadvantages of the features are discussed. The investigation of these features gives an opportunity to evaluate the optimisation approach.

This thesis contributes increased knowledge of the flow pulsation created due to unsteady flow through a valve plate opening. This increases the simulation model’s accuracy.

1.4 Delimitation

Only the hydraulic motor/pump unit is considered. Hence, the external system has been ignored as far as possible. In this way the hydraulic machine can be investigated and improved independently of the external hydraulic system. The machine is most often designed without knowledge of the external system. However, the hydraulic system noise creation is not less important and the reduction of noise in a hydraulic motor/pump unit can vanish in a badly designed system.

Axial piston machines are used during dynamic behaviour analysis in the thesis. This kind of machine is usually used in high pressure applications where the noise problem is most obvious. The theories and conclusions are considered to be valid for other positive displacement machines too. Further tests and understanding of the transfer function methodology are needed to verify this, also for an in-line axial piston machine.

The thesis considers only characteristics which are judged to create noise and vibration. Other important design aspects such as efficiency and controllability are therefore not treated.
1.5 Outline

This thesis builds on the work in the appended papers. The next chapter contains a more in-depth background to the papers and the work performed. However the papers form the bulk of the thesis and for deeper understanding of the work and results the reader is referred to the appended papers.

The first chapter is an introduction to the thesis followed by a review of the fundamentals regarding noise in hydraulic machines and a state-of-the-art review of the subject. Simulation, optimisation, source flow and noise measurement have been an important part of the thesis and one chapter is allocated to each of these topics. The noise reduction interventions which are performed are described immediately before the conclusions and outlook chapters. The appended papers can be found at the end of the thesis.
Noise in fluid power systems

Noise generated from mechanical vibration can be explained in three steps: **Exciter → Resonance answer → Noise emission**, according to [11]. The course of events generating noise starts when a force, i.e. exciter, touches a surface and depending on the geometry, stiffness, and damping, the force creates vibrations. If the force touch is rapid the noise is of higher frequency compared to a slower touch. This can be translated to a hydraulic axial piston machine as: Exciter is the piston which creates both a flow ripple and oscillating forces. The pump casing and connected structure are the resonance structure. If the fluid in the machine is rapidly pressurised, the sound will be loud at higher frequencies compared to a more gradual pressurisation of the fluid. The machine housing and connected pipes and hoses affect the resonance answer of the machine. The amount of emitted noise depends on the size of the surface, the boundary layer between the surface and the surrounding air, and the nature of the vibrations.

The perceived noise level depends on the frequency of the pressure waves in the air, not only the measured sound level. Along a so called isophone curve, the perceived noise level is constant for different frequencies, despite the measured noise level not being constant, [11]. The isophone curves have slightly different shapes at different perceived noise levels. Higher frequencies are however more disturbing than low frequencies with a peak between $\approx 2 - 4 \text{ kHz}$. Different acoustic weighting filters are typically used to compensate for the frequency dependent perception.

There are three types of noise in a hydraulic system: **Fluid borne noise → Structure borne noise → Air borne noise**. The fluid borne noise is generated by a pressure ripple which is created when flow pulsation interacts with the system’s impedance. The pressure ripple interacts with the pipes
and the structure to which the pipes are connected and produces dynamic movements of the mechanical structure, generating structure borne noise. This kind of noise is also created directly from forces and bending moments which interact with the structure. When the vibrations interact with the surrounding air, the air starts to vibrate and air borne noise is created.

2.1 Noise at its origin

The noise in a hydraulic system is considered to origin from the fluid power pumps and motors which are exclusively of displacement type. It is important to consider both structure borne noise which is mainly attributable to the internal forces and moments in the machine and fluid borne noise which originates in flow pulsations at valve plates or equivalent.

Many different types of displacement machines exist, [49] e.g. vane machine, gear machine, radial piston machine, axial piston machine etc. The working principle is the same for all kinds; the displacement machine divides the inlet flow in chambers and then delivers fluid to the outlet line. Axial piston machines, which are used exclusively in this thesis, are divided into two groups: Bent axis machine and in-line machine or swash plate, figure 2.1. The displacement in a bent axis machine is achieved by an angle between the barrel’s rotating axis and the drive shaft while the displacement in the in-line machines is achieved by titling the swash plate from a plane perpendicular to the rotating axis.

2.1.1 Valve plates

One of the most important features of an axial piston machine is the valve plate, which is used to separate the inlet from the outlet ports. The valve plate can easily be modified to reduce noise, both the flow ripples and force pulsations, in pumps and motors. The figures in 2.2 show typical configurations of valve plates used for research purposes and commercial machines. Figure 2.2(a) shows a zero-lapped valve plate which means that immediately a cylinder leaves one port it connects to the other. An easy and effective way to reduce the flow ripple is to add a pre-compression angle, figure 2.2(b). The pre-compression angle is very sensitive to different operation points. These two valve plate types are more or less only used for research purposes. Figure 2.2(c) shows a typical valve plate mounted in a hydraulic pump, i.e. the machine works with unchanged rotation direction and unshifted pressure port. The valve plate has a pressure relief groove at the pre-compression angle, which makes pressurisation smoother and less sensitive to changing operating conditions. The most used design of the pressure relief groove is a triangular shaped groove with a linear increase of the height, \( h \). This is mainly used for production reasons. The valve plate in figure 2.2(d) has pressure relief grooves at all four port angles. This valve plate is used in machines working with different rotation direction and/or
Noise in fluid power systems

Figure 2.1 Cross-sectional illustration of axial piston machine. Courtesy of Parker Hannifin.
switching pressure ports. This valve plate is referred to as a motor valve plate. These four valve plate configurations are frequently used in the thesis and are referred as zero-lapped, pre-compression angle, pump, and motor valve plate.

Figure 2.2 (a) and (b) are two typical valve plate configurations for an axial piston machine used mainly for research purposes. (c) and (d) show two typical valve plate configurations used in commercial axial piston machines.

2.1.2 Flow pulsations

The machine delivers a mean flow with a superimposed flow ripple caused by the discrete number of pumping elements, i.e. kinematic flow pulsations, and the limited stiffness of the oil, i.e. compressible flow pulsations. In the method used in [95] and section 6.1 it is shown that the structure vibrations and audible noise at the pump housing can be predicted with high accuracy with only the simulated forces created inside the machine; the flow pulsation noise contribution at the pump housing is thus of minor significance. However, the overall audible noise contributions to a hydraulic system, the flow pulsation is of great importance [66].
Kinematic flow pulsation

The total kinematic flow pulsation at the outlet is normally determined as the sum of all positive flow contributions from each cylinder, equation (2.1) together with equation (2.2).

\[ q_k(t) = \varepsilon D_n \sum_{j=1}^{z} g_j(t) \quad g_j(t) \geq 0 \]  

(2.1)

where

\[ g_j(t) = \sin \left( 2\pi nt + (j - 1) \frac{2\pi}{z} \right) \]  

(2.2)

The kinematic flow pulsation can be seen in figure 2.3(a) as the solid line ripple. The Fourier series expansion of equation (2.1) is shown in equation (2.3) for odd and even piston numbers, [88]. The fundamental harmonic is located at a frequency of \( \omega_0 = 2\pi zn \) if the pistons are evenly distributed in the cylinder barrel and drive shaft. The rest of the harmonic frequencies are multiples of the first harmonic. In the case of non-uniform placement of the pistons, as in paper [IV], the fundamental frequency is only dependent on the rotational speed and not the piston number, \( \omega = 2\pi n \).

The odd piston machines are preferred with regard to flow ripple where only every second harmonic contains energy compared to even piston numbers.

\[ Q_k = 2\varepsilon D_n \frac{1}{(zm)^2 - 1} \quad \left\{ \begin{array}{ll} m = 2, 4, 6, \ldots & \text{for odd piston numbers} \\ m = 1, 2, 3, \ldots & \text{for even piston numbers} \end{array} \right. \]  

(2.3)

Another way to illuminate the difference between odd and even piston numbers is to look at the degree of non-uniformity of the kinematic pulsations in the time domain as it is done in [88], equation (2.4). Since the pistons enter and leave the ports in pairs for a machine with even piston numbers, the uniformity becomes greater compared to an odd piston machine where the cylinders enter and leave the ports alternately with a half period separation.

\[ \delta_k = \frac{\Delta q_k}{q_p} = \upsilon \frac{\pi^2}{z} \quad \left\{ \begin{array}{ll} \upsilon = \frac{1}{8} & \text{for odd piston numbers} \\ \upsilon = \frac{1}{2} & \text{for even piston numbers} \end{array} \right. \]  

(2.4)

To reduce the kinematic flow pulsation, the most obvious way is to add pistons in pairs, i.e. keep the odd piston number. Changing the geometrical motion of the pumping elements is another way to change the kinematic contribution, which is done for a radial piston machine in [87]. Reducing the kinematic contribution is more valuable for a machine with low compressible pulsations like gear pumps and machines working at low pressures. In paper [IV], the pistons’ location and cylinder kidney angular position was optimised to reduce
the overall flow pulsations and structure borne noise at the machine housing. A similar approach is taken in [108] for a vane pump, but only the maximum flow pulsation harmonic was minimised.

**Compressible flow pulsation**

The compressible part of the total flow pulsation is caused by a non-matching cylinder pressure when the cylinder connects to the high or low pressure port. The compressible flow is a sudden flow going into or out from the cylinder. Most of the research contributions are about the compressible part of the flow pulsation, mainly because it is the most important cause of high flow pulsations, see the dashed line and empty bars in figure 2.3. In addition, the compressible part of the flow pulsation is fairly easy to change in form and in this way reduce the energy content.

In analogy to the degree of kinematic non-uniformity, the same can be done for the compressible flow pulsations [88]. The grade of compressible non-uniformity can be expressed as equation (2.5).

$$\delta_c = \frac{\Delta q_c}{q_p} = \left(1 + \frac{n}{2}\right) \frac{\alpha_p p_x}{\tau \beta_c}$$  \hspace{1cm} (2.5)

The parameter $n$ is the area exponent and can for most common used opening geometries, be expressed as equation (2.6).  

$$A = K \Phi_c^n \text{ where } n = 0, 1, 1.5, 2$$  \hspace{1cm} (2.6)

$n=1$ denotes a linear increasing area and $n=2$ the common used triangular groove. $n=1.5$ denotes the opening created by the intersection area of two circles, i.e. the case when no groove is used.
The parameter \( \alpha_p \) is the relative dead volume in the cylinder volumes. In general, the parameter is bigger for in-line machines compared to bent-axis machines. The parameter \( \tau \) is the relative charging time, i.e. the relationship between the time where there is a back flow into the cylinder due to the oil’s compressibility and the period time as equation (2.7). \( \tau \) is dependent on many pump parameters, e.g. displacement, cylinder dead volume, the valve plate design, number of pistons, and system pressure.

\[
T = \frac{2\pi}{\omega_p z}
\]  

(2.7)

The parameter \( \tau \) is easy to determined by the geometry of the pump and the air content in the oil. The parameter is normally between 0.1 and 0.3. The larger value associates with a quiet pump with pressure relief grooves at the connection to the ports. As can be seen, the non-uniformity is directly proportional to the system pressure and inverse proportional to \( \tau \). For a commercial five-piston pump with a zero-lapped valve plate at high outlet pressure, the compressible non-uniformity can be calculated to 0.5 while for the same pump the kinematic non-uniformity is 0.05. This example is for an extreme case, but the compressible non-uniformity is generally higher than the kinematic non-uniformity in systems with high pressure. It is obvious to make the greatest effort for the compressible pulsations. By contrast, other type of pumps such as vane pumps where the compressible pulsations are less dominant, research on kinematic pulsations is more legalised.

This theoretical formulation is valid for constant bulk modulus. Thus all the simulations and measurements are made with boost inlet pressure to reduce the bulk-modulus dependency.

As shown in [88], compressible flow pulsation spectrum is mainly influenced by \( \tau \) and to a limited extent by how the valve plate opening is made. This is highly connected and shows the importance of having a very smooth pressurisation of the cylinder fluid.

It can be shown that the relative amplitude, equation (2.8), of the first harmonic is almost unchanged by different values of \( \tau \) but the higher frequencies are strongly dependent on the relative charging time.

\[
\frac{Q_1}{Q_p} = 2\alpha_p \frac{p_0}{\beta_c}
\]  

(2.8)

Pre-compression filter volume is one of few examples where the amplitude of the first harmonic can be reduced.

### 2.1.3 Internal forces and moments

Pulsating forces are created inside the machine when the cylinders link up to the high or low pressure ports and also because of the rotating barrel which is
moving the axial piston forces’ point of action. The axial piston force for one cylinder is calculated according to equation (2.9). The point of action of the axial piston force according to the coordinate system in figure 2.4 is shown in equations (2.10) and (2.11).

\[ F_c = A p p_c \]  
\[ x_c = R_b \sin(\phi_c) \]  
\[ y_c = R_b \cos(\phi_c) \]

The total axial force is the sum of all the cylinder forces, equation (2.12). The force is applied on the valve plate.

\[ F_z = A p \sum_{j=1}^{z/2} p_{c,j} \] 

In an odd piston number pump with ideal zero-lapped valve plate, the number of cylinders connected to the high and low pressure ports alternate between \( z/2 + 0.5 \) and \( z/2 - 0.5 \). The resulting axial piston force for a zero-lapped valve plate becomes a pulse train because of the rapid pressurisation in the cylinder. This is the most undesirable profile in a noise perspective due to the energy.
A pump with an even piston number with ideal zero-lapped valve plate, the number of cylinders connected to the high and low pressure kidneys is always \( z/2 \) and the total axial piston force profile is constant except for a small, rapid transient when the cylinders are connecting to their respective port. This is the most common argument for choosing even piston number pumps; however, the flow pulsation is heavily increased as stated earlier. The even piston numbers can be preferable in some systems, i.e. a system with large flow pulsation damping and large vibrating surfaces connected to the pump housing. Considering the amount of odd piston number compared to even piston number pumps on the market, the flow pulsation is far more important than the structure borne noise created by the axial piston force.

The moment created by the total axial piston force is calculated according to equation (2.13) and (2.14). The point of action for the total axial piston force can be written as equations (2.15) and (2.16). This is shown as an uneven butterfly in figure 2.4.

\[ M_x = \sum_{j=1}^{z} F_{c,j} y_{c,j} \]  
\[ M_y = \sum_{j=1}^{z} F'_{c,j} x_{c,j} \]  
\[ x = \frac{M_y}{F_z} \]  
\[ y = \frac{M_x}{F_z} \]

A comprehensive discussion about piston forces and moment and their importance of noise contributions can be found in [53]. The conclusion of the reference is that a sinusoidal varying force profile is the most preferable profile not only because of the minimised oscillation energy but also the bending moments and drive shaft torques become almost sinusoidal and collected to one harmonic, which makes it easier to reduce the noise contribution from this frequency by a correct pump housing design. Paper [II] shows that the best all-round objective function is the derivative of the cylinder pressure which implies the smoothest possible force profile.

The axial piston force and its created moments are the main source of structure vibration at the pump housing. This is elucidated in [95] where the authors show that the audible noise and structure vibrations can be predicted with high accuracy by just the total axial piston force and the bending moments at the valve plate. The experimental work shows that the drive shaft moments are of minor importance; however, the application for which the investigation is made probably has high influence on this statement. In section 6.1 further measurements are made with a slightly different pump design, but the same concluding remarks can be made. The measurements also show that the total
axial piston force has greater impact on the noise and vibrations at the pump housing than bending moments, but the bending moments at the valve plate are nevertheless not negligible.

2.2 Early research contributions

The study of noise reduction in hydraulic systems began more than 40 years ago. One early scientific work was Helgestad's PhD thesis in 1967, [41]. In the same year Simpson [101] published a theoretical investigation of hydraulic noise generated in a pump. Another early work on noise reduction was performed in Russia and published in 1969 [125]. Helgestad continued his work in [42, 43] where the importance of pressurisation of the cylinder volume were presented. Ideal timing and the pressure relief groove's effects were presented as well as an idea for using check valves to obtain ideal timing.

In the mid-1970s, a major study was conducted in the United Kingdom in a project called Quieter Oil Hydraulic Systems. The University of Bath is one of the most prominent universities in the field and began its era during the project.

Several different flow ripple measurement methods were developed in the late 1970s and the beginning of the 1980s. Hydraulic trombone [21] and the high impedance method, which was also adopted by British Standard Institution, [80] were two early measurement methods. These methods were developed at the University of Bath. Another measurement method of flow ripple from hydraulic pumps is the anechoic termination method, [13, 73]. In the early 1990s two different methods were developed by the University of Bath and Linköping University, viz. the secondary source method [24, 25] and the two-microphone method [92, 112] respectively.

In parallel with the early development of measurement methods in the 1970s, researchers looked at how the flow ripple interacts with the external system and the creation of pressure ripple, see [81] among many others. Another early track is structure vibration investigations [19, 36], where focus is on transformation from pressure ripple to vibrations in pipes and hoses. The work on noise reduction from pipes and hoses has continued since then, a good summary of fluid transmission line models can be found in [103, 104] while in e.g. [71, 78], the focus is on noise reduction and pressure pulsations in hoses.

In [72], different types of passive dampers were investigated and tested. This was the first licentiate thesis in the noise reduction area at Linköping University and since then the university has produced three PhD theses in the area, [53, 92, 113] and in addition some closely related works have been produced. A more comprehensive summary of early research contributions on noise reduction until 1999 can be found in [20].
2.3 State of the art in noise reduction in piston machines

One simple way to reduce compressible flow ripple is to delay the kidney opening angle, i.e. the pre- and decompression angles. The angles can be optimised for a specific displacement angle and pressure level but if the conditions are changed the angles are no longer optimal and the design can even increase the noise level for certain conditions. A detailed study of the pre-compression angle is presented in [27].

The compression part can be modified with different techniques, the most common in commercial pumps/motors being pressure relief groove. A dimensionless study of optimal design of pressure relief grooves is presented in [88].

A satisfactory pre-compression can be accomplished by a check valve implementation at the pre-compression angle. The opening angle to the kidney is delayed so much that the cylinder volume is sufficiently compressed over the whole operation pressure range. The check valve is opened when the cylinder pressure rate reaches the discharge level. The check valve was tested in [32]. According to Grahl, fatigue is the method’s main problem.

One idea based on the check valve function is the vortex diode which was tested in [116]. The feature was not, however, suited to the application in question due to the low dynamic performance. In [35], several highly damped check valves were mounted in series. The valves are stationary opened due to the pressure balance over the poppet and when the condition changes the valve closes. The design prevents the oscillating behaviour that appears with ordinary check valves.

Another feature is the pre-compression filter volume (PCFV). The feature contains a volume which is linked up to the cylinder before connection to the high pressure port. The pressure in the volume enables a smooth pressure build up in the cylinder. The volume is recharged with a small amount of flow from the high pressure port before the volume is released from the port and cylinder. PCFV was invented at Linköping University in the early 1990s [93, 94, 115]. In these references the flow pulsations were in focus and the suggested volume size is three times the size of the displacement chamber. In [117] and [98], it is pointed out that the pressurisation rate is increased and hence the directly emitted noise is also increased. However, paper [III] shows when the audible noise is minimised with the use of pre-compression filter volume. The volume size should be bigger to minimise the audible noise compared to flow pulsations. Furthermore, the flow pulsation and audible noise can be reduced more compared to an ordinary pressure relief groove. The PCFV was patented in 1993 [89].

In [34], the PCFV is used at the entrance to the inlet kidney and is called pre-expansion volume. The volume is used to reduce the cavitation, and hence it may be possible to increase the self priming speed. In [85] an actively controlled

\[\text{The feature is also commonly called ripple chamber and pre-compression volume.}\]
PCFV is used to reduce the pulsations in an axial piston pump/motor; the flow between the volume and cylinder is controlled by ideally dynamic control valve.

Grahl investigated the possibility to obtain optimal port plate timing with various designs [32]. Rotating valve plate is introduced in the reference and through on-line adjustments optimal port plate timing is conceivable. The design is expensive due to the on-line control devices and also expensive implementations. However, as the span of operational conditions and the environmental requirements concerning noise increase, industry is being forced to consider more expensive solutions, Grahl claims in [32]. More recent attempts to have active control of the pre-compression can be found in [86]; an actively controlled valve plate is investigated and significant reduction of pressure ripple were found.

A so called cross-angle which is a fixed angle perpendicular to the normal displacement angle in variable pumps/motors. The effect of the cross-angle is that the top and bottom dead centres move with the displacement setting of the machine. Thus, it could in some ways be compared to a revolving valve plate, but with less flexibility as the cross-angle is fixed. However, with the cross-angle it is possible to make the pulsations less sensitive to variations in the displacement angles of the machine. The cross-angle is primary used in constant pressure systems but it can also be used together with other noise reduction features, sections 7.2. The cross-angle was first patented by Citroën [17] and appeared in literature eleven years later in 1974, [7, 43]. The cross-angle has been exhaustively investigated in [54, 55], and in [56] the cross-angle was experimentally tested with good results. In paper [I] and section 7.2, the cross-angle was investigated for motors as well as motor/pump applications. In [40] an actively controlled cross-angle using a piezo-actuator is developed. The simulation results show a slightly reduced flow pulsation; however, the authors have limited prospects for using piezo-actuator due to the large piston forces on the swash-plate.

In general, there are many different concepts which are patented and investigated to adapt the noise reduction features to certain operating conditions in the fluid power system, in an active or passive way. However, due to the high frequency of the opening and closing of the outlet and inlet ports in a traditional axial piston pump/motor, i.e. valve plate configuration, the concepts will probably not be suitable for fluid power machines with long lives. Also, most of the concepts are expensive and the reliability is questioned. The manufacturers are not mature enough to take a big revolutionary step in machine improvement. However, the improvements in other areas; competing electrical systems, reduction of noise from other sources around where the fluid power systems are placed such as combustion engines, the fluid power community has to rethink their development strategy. Also, the fluid power systems show possibilities in new markets, such as hybrid passenger cars, where the noise problem has higher priority compared to construction machinery.

\[^2\]The feature is also called secondary swash plate angle.
2.4 New pump/motor concepts

A number of new design concepts have been developed which may decrease the noise level in hydraulic machines but some of the concepts may increase the force and flow pulsations created in the pumps/motors. A pump developed by Artemis is presented in [28] where two separate valves are connected to each cylinder at the high and low pressure ports. The valves can be controlled individually and opened when the cylinder pressure is the same as in the discharge kidney. Also, the pump is variable since the cylinders are connected to the low pressure kidney through the whole rotation. In this way the pump does not deliver any oil and thus no work has been carried out. The manufacturer claims that the new technique leads to an increase in efficiency compared to the original variable piston pump/motor, [91]. The way of controlling the displacement may significantly increase the noise level due to the increase in kinematic flow pulsations.

A new type of axial piston machine was presented in [3], Innas Hydraulic Transformer concept, IHT, which has three ports on the valve plate: A-port, B-port, and tank port. The design is called “floating cup”. The principle is mainly designed as a hydraulic transformer, but the design also has potential for a pump/motor concept, [1] and variable machines, [2]. The bifurcate machine have a total of 24 pistons, 2x12 = 24, and due to the high amount of pistons the machine is relatively quiet with a high frequency noise. The benefits and disadvantages of having two connected pump halves are investigated in [107].

In [57], the shuttle technique is explained for an IHT-machine where big pressure peaks are expected. The method reduces the pressure peaks in a simulation environment and considered hence to reduce structure borne noise. The technique is expected to have several other advantages.

Artemis, mentioned earlier, is the leading developer in digital pump/motor applications. In [77], another hydraulic digital machine is presented. [76] considers the switching valves as an potential source of noise for digital pumps and motors. In [83], a digital pump/motor concept shows promising results as regards efficiency but as the authors state, the noise may be a problem and further investigations and improvements are needed for digital hydraulic pump and motor units.

2.5 Multi-quadrant pump/motor

Most research concerns pump application, which is natural because of the amount of pumps compared to motor applications. In general, every hydraulic system needs to have a pump application regardless of how the transformation from hydraulic power to mechanical power is made, e.g. motor, cylinder etc. Hence, there are more pumps than motors on the market. Also, the trend in hydraulic development is to make the machines more flexible in the sense of combined motor and pump modes, for example to use them in secondary
control strategies to enable energy recovery or in vehicle transmissions. Machines with controlled displacement angle to reduce the power losses are also increasing in number.

A machine can have eight different driving modes, so called quadrants; see figure 2.5. \( n \) on the y-axis is the rotational speed where positive values have an anticlockwise rotation and negative values a clockwise rotation. The displacement angle on the x-axis can be both positive and negative and the pressure port can be swapped. All these driving modes need different pre- and decompression angles. The principle design of the valve plates for the different driving modes is also shown in figure 2.5. It is obvious that it is difficult to find a suitable valve plate for all driving modes without getting cavitation, flow ripple and pressure overshoots. A good, simple compromise is pressure relief grooves or comparable features. The optimal solution would be to have free pre- and decompression angles; thus, for every driving condition, the angles are tuned for the lowest possible noise level. This may not be at all practicable and if so the solution is considered expensive. Artimes’ cylinder opening solution with a separate digital valve at each cylinder can be seen as free pre- and decompression angles, [28]. Revolving valve plate [32], which enable changing position of the pre- and decompression zones, is a compromise for free pre- and decompression angles. The feature has limited adaptation due to the angles can not be changed individually.

The different noise reduction features available are more useful in some applications than others depending on how the pump/motor unit is supposed to work and in which system the unit is placed.

**Variable pump and motor**

One of the most extreme quadrant shifts for the valve plate design is when the displacement angle is turned over zero, i.e. both positive and negative displacement angles. This is due to the location of the pre- and decompression angles, see figure 2.5. However, by using a cross-angle the problem is reduced due to the displacement of the cylinder dead points, hence the pre- and decompression angles are changed. In all variable pump and motor units with fixed pressure ports it is preferable to use cross-angle. For swash-plate units the implementation is very simple and is already used in some smaller pump applications. Bent-axis implementation is probably more adverse due to the moveable cylinder barrel.

**Pressure ports**

Shifted pressure port is another extreme case; the location of the pre- and decompression angles are changed, see figure 2.5. There is no particularly good, simple solution for this quadrant shift. If an optimisation of the pre- and decompression angles are performed for shifted pressure ports simultaneously, the results become very similar to a zero-lapped valve plate, i.e. neither pre-compression nor decompression angles are preferable. However, use of pressure relief grooves with fairly large opening
Figure 2.5 The different valve plates show how the pre- and decompression angles should be designed for different driving modes. The angles are merely schematic. Positive values of $n$ denote anticlockwise rotational direction in the figures.
areas are preferable to a zero-lapped valve plate. More extreme solutions for this quadrant change might be variable cross-angle or rotating valve plate.

**Constant pressure pump/motor**
Units which are working with constant pressure during operation the most useful feature is ideal timing i.e. fixed pre- and decompression angles.

**Variable pressure pump/motor**
If the pump works with different pressure levels the pressurisation has to adapt to the pressure level in the system. The simplest feature is the pressure relief grooves and is suitable for most pump and motor units. Pre-compression filter volume is more advanced to implement and also most likely more expensive. However, for more advanced pump and motor units and systems, the feature is justifiable due to the improved adaptation for different operating conditions such as pressures compared to pressure relief grooves.

**Rotation direction**
The change of rotation direction does not make any big difference to the pressurisation zones. The pre-compression zone is changed to a decompression zone and vice versa and only small changes of the pre- and decompression angles are needed. In addition, the pressure relief grooves are minor, depending on the rotation speed, and pre- and decompression angles are almost not affected.

**Motor units**
In general, the motor mode is difficult to improve for all frequencies, as is explained in section 2.5.2. The same assumption as for pump modes is valid for the motor modes.

Table 2.1 shows how various systems and noise reduction features in pump/motor units should be combined. The table shows which kind of features in the pump/motor units are most useful for the various systems. In the table, the gradual pressurisation feature may be pressure relief grooves or pre-compression filter volumes or other similar features.

**2.5.1 The machines’ system dependency**
The thesis focuses on the machine itself and the external system is eliminated. This is because it is not evident in what kind of system the pump/motor unit will be used when the machine is made. The basic idea of eliminating the system dependency is that if the source of noise, i.e. pump and motor units, emits less noise the hydraulic system will probably produce less noise too. However, a quiet pump/motor may become very noisy if the external system is badly designed.
Table 2.1 *The table shows how various systems and noise reduction features in pump/motor units should be combined.*

<table>
<thead>
<tr>
<th></th>
<th>Cross-angle</th>
<th>Gradual pressurisation</th>
<th>Fix pre-compression angle</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>OPEN SYSTEMS</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Closed-centre system with fixed pump and pressure relief valve&lt;sup&gt;a&lt;/sup&gt;</td>
<td></td>
<td></td>
<td>x</td>
</tr>
<tr>
<td>Open-centre system with fixed pump</td>
<td>x</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Load sensing system with fixed pump and bleed-off valve</td>
<td>x</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Closed-centre system with variable pump</td>
<td>x&lt;sup&gt;b&lt;/sup&gt;</td>
<td>x&lt;sup&gt;b&lt;/sup&gt;</td>
<td></td>
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<tr>
<td>Load sensing system with variable pump</td>
<td>x&lt;sup&gt;b&lt;/sup&gt;</td>
<td>x&lt;sup&gt;b&lt;/sup&gt;</td>
<td></td>
</tr>
<tr>
<td>Secondary control system with variable pump/motor</td>
<td>x&lt;sup&gt;b&lt;/sup&gt;</td>
<td>x&lt;sup&gt;b&lt;/sup&gt;</td>
<td></td>
</tr>
<tr>
<td><strong>CLOSED SYSTEMS</strong></td>
<td></td>
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<td></td>
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<tr>
<td>Closed transmission</td>
<td></td>
<td></td>
<td>x&lt;sup&gt;c&lt;/sup&gt;</td>
</tr>
</tbody>
</table>

<sup>a</sup> The system is only used at very low effects.  
<sup>b</sup> The features will be used to complement each other.  
<sup>c</sup> Very limited usability.
Structure vibrations are important in some systems while flow ripple is the major cause of noise in others. If the machine is connected to a short pipeline system or if the pipes are connected to large volumes, the flow ripple is most probably less important compared to structure vibrations.

The opposite case is in many industrial applications where the pumps are placed in a separate room and only flow pulsation is exposed to the surroundings. In such situations the flow pulsation is highly important while structure vibration at the pump housing is less prominent.

A typical noise reduction feature where the placement of the machine is important is the pre-compression filter volume. Hence, a small pre-compression filter volume may be better to minimise the flow pulsations while a larger volume is preferable when the emitted noise level at the pump housing should be minimised. Figure 2.6 shows the trade-off between the audible noise level at the pump housing and the flow pulsation. Both optimisation of pressure relief groove and pre-compression filter volume are shown. The figure shows the increased reduction of both flow pulsations and audible noise. However, the audible noise and flow pulsation can not be minimised simultaneously but both objectives can be reduced. Figure 2.7 shows the corresponding volume size and the volume becomes significantly greater when the audible noise at the pump housing is minimised. Paper [III] shows further results from these optimisations.

![Figure 2.6](image)

**Figure 2.6** The trade-off between audible noise and flow pulsation when pre-compression filter volume is optimised, shown as points. This is compared to optimisation of pressure relief groove, shown as crosses. The dotted oval marked with 1 and 2 connects to the ovals in figure 2.7.

### 2.5.2 Motor mode

Most of the literature is solely about hydraulic pumps; however, the conclusions are also generally valid for motor modes. In [47] a bent-axis motor is
investigated and from the results the pressure ripple generated in motors can be compared to the levels in pumps. Paper [I] investigates the possibility to use a cross-angle in a machine with both positive and negative displacement, i.e. a machine working both as pump and motor but with unshifted high and low pressure ports. Further discussion of the cross-angle in different driving condition can be found in section 7.2.

Paper [III] elucidates the difference between motor and pump units; one of the statements is that motors produce less audible noise but the flow ripple and emitted noise are still significant. One possible reason for the lower noise emissions are that the pre-compression angle is always located at the top dead centre where the cylinder volume is small. However, the flow pulsation in the low pressure port is in the same size as the high pressure port.

A significant difference between motor and pump mode is the realisation of pre- and decompression angle. Figure 2.8 shows the typical pre- and decompression angles for pump and motor mode when the rotation direction is changed. Pre-compression angle is used to reduce the compression flow pulsation since the output flow from the cylinder is delayed the kinematic flow pulsation is increased. This is not a problem in the pump mode due to the reduction of the compressible flow pulsation being greater than the kinematic flow pulsation increase and both occur in the same port, i.e. the high pressure opening is delayed to reduce the compression flow pulsation in the same high pressure port. On the other hand, for the motor mode, to reduce the compression flow pulsation in one port the opposite kidney has to be modified. This results in an increase in kinematic pulsation in one port when the purpose is to decrease the compression pulsation in the other.

This is also illustrated in figure 2.9 where the flow pulsations in high and low pressure ports with a zero-lapped valve plate are compared to a valve plate with pre-compression angles. The valve plates are optimised to minimise the high and low pressure port flow pulsation in the time domain simultaneously.

The flow pulsation in the high pressure port is drastically increased for the
fundamental frequency in the motor mode simulation, figure 2.9(a), while the inlet flow pulsation is reduced, figure 2.9(c). In the pump case shown in figure 2.9(a), the pulsation is reduced in both high and low pressure ports, albeit very little in the low pressure port. Note that the valve plate configuration is not the same for the pump and motor simulations.
Figure 2.9 The flow ripple in both the time and frequency domains in the high and low pressure port at maximum positive and negative displacement angles. Dashed lines and bars are zero-lapped valve plate and solid lines and points are pre-compression angle valve plate.
Simulation

Simulation techniques are valuable tools in the early stages of the development process. Detecting a fault early in the design phase is many times cheaper than later in the development stage [6]. The design freedom is big in the beginning of the design phase but knowledge of the product and its design problem is limited. By simulating the product in an early state, knowledge is enhanced and the design freedom can be used to improve the product from scratch. In the later stage, small improvements can be made to increase the product’s excellence. The simulation model can be used to gain better understanding of the product. With simulation, characteristics which are difficult to measure in physical measurements can be supervised in a simulation environment.

The first question to ask when building a simulation model is: “What is the purpose of the simulations and what do I have to include in the model to create a sufficient model of the purpose?” The simulation model should include the information which is important for the purpose of the simulation, neither more nor less.

For system simulations, it may only be of interest to simulate a typical pump flow ripple to understand how the external system interacts with the pump-excited energy. In [58], the fluid power pump flow spectrum is simulated with only two parameters with good results which is sufficient for most system simulations. This information is not enough for a pump designer. For this purpose, a more comprehensive model has to be used with higher complexity. The simulation model needs to capture how certain dynamic characteristics of the pump change when design modifications in the pump are made. Over the years many different pump models are developed in different simulation environments for this purpose, see for instance [30, 79, 97], CASPER [50, 99, 120], HASP [22, 46].

In this thesis the one-dimensional (1D) comprehensive simulation model is created in the simulation program HOPSAN [44, 119], further explained in
To study the noise excited in a fluid power machine; flow pulsations and internal force are considered to be the main output characteristics. The most important parameters for noise investigations in hydraulic pumps are the timing of the valve plate, pressure, rotational speed, and fraction of displacement when a variable displacement pump is considered.

It is appropriate to use Computational Fluid Dynamics (CFD) when analysing the design and performance of the fluid power machine due to the dependency of the fluid hydrodynamics. Compared to the 1D model, a more detailed view of the fluid in pipes, orifices, and cylinders etc. can be visualised with a three dimensional (3D) CFD model. Compared to experiment, the flow can be studied in detail without any change of the hardware which is many times very difficult or impossible. The flow field inside the pump is studied in [121] and shows the usefulness of CFD simulations. In [124], the CFD software Fluent is used to investigate the pump dynamics in an axial piston pump. CFD techniques have made good progress in numerical stability and also increasing speed of computers in recent years. CFD software becomes an important tool for pump designers which can be seen in the increase in research papers considering CFD simulation of axial piston machines. In [38], an operation similar to an optimisation is performed together with CFD software and several valve plate configurations were simulated during one night. Another example is [8] where a Helmholtz resonator is used to improve suction port behaviour by means of 3D CFD simulations.

In this thesis the 3D CFD model is made in PumpLinx from Simerics, Inc [18], further explained in section 3.3. The CFD model gives a wider and more detailed view of the flow inside the machine but is many times slower than the 1D model. The 1D model is 200,000 times faster compared to the CFD model used in this thesis, despite the CFD program being fast compared to other CFD programs. The CFD model can rarely be used together with an optimisation algorithm where several iterations are needed, however, the utilized amount of information is much higher compared to the 1D model. The question that should be asked is again: “What is the purpose of the simulations and what do I have to include in the model to create a sufficient model of the purpose?”

3.1 Hopsan

The comprehensive simulation model is implemented in the program HOPSAN. The first version of the program was developed at Linköping University in the late 70s. A new generation of the simulation program was released in 2011 [29]. In HOPSAN the model is structured in components which are connected by power ports. For the connections transmission line theory (TLM) is utilised [69], thus the components are numerically separated and local solvers can be deployed. This is backed by the physical effect of limited signal propagation velocity and is advantageous for stiff systems.
The pump model used in this paper originates from [114], further developed in [119]. The model is modularised and is easy to change in terms of different pump and motor geometries. The machines investigated in this study are axial piston machines of both in-line and bent-axis type but the modelling principle also works for different displacement machines such as vane pumps and radial piston pumps. The pump model is distributed to individual components, i.e. each cylinder in the pump, two orifices per cylinder representing opening to high and low pressure port respectively, etc. In this way the system is kept close to reality. This is illustrated in figure 3.1. It is easy to add additional features to the pump, such as pre-compression filter volume.

![Figure 3.1 Illustration of the distributed pump model structure where each component in the pump has its own model component with the required equations.](image)

Figure 3.2 illustrates how the flow is modelled in Hopsan. The cylinder capacity is modelled as a volume, dependent on the cylinder volume and bulk modulus. The cylinder capacity represents the compressible flow ripple, equation (3.1).

\[
q_c = \frac{V_{cyl} \, dp}{\beta_c \, dt} \tag{3.1}
\]

The term \( \frac{dp}{dt} \) is the pressure change inside the cylinder. \( V_{cyl} \) is the cylinder

![Figure 3.2 Illustration of the flow modelling in Hopsan. Kinematic flow modelled by piston motion and compressible flow modelled with a capacitance.](image)
volume which changes during the rotation according to equation (3.2) and (3.3) for bent axis and in-line machines respectively.

\[
V_{\text{cyl}} = V_0 + R_d \pi R_p^2 \left( \frac{2 \sqrt{\sin^2 \alpha + \sin^2 \gamma}}{2} + (\sin \varepsilon \sin \phi_c + \sin \gamma \cos \phi_c) \right) \tag{3.2}
\]

\[
V_{\text{cyl}} = V_0 + R_b \pi R_p^2 \left( \frac{2 \sqrt{\tan^2 \alpha + \tan^2 \gamma}}{2} + (\tan \varepsilon \sin \phi_c + \tan \gamma \cos \phi_c) \right) \tag{3.3}
\]

The kinematic flow is modelled as the derivative of the cylinder volume, equation (3.4) and (3.5).

\[
q_k = R_d \pi R_p^2 \left( \sin \varepsilon \cos \phi_c \frac{d\phi_c}{dt} + \sin \gamma \sin \phi_c \frac{d\phi_c}{dt} \right) \tag{3.4}
\]

\[
q_k = R_b \pi R_p^2 \left( \tan \varepsilon \cos \phi_c \frac{d\phi_c}{dt} + \tan \gamma \sin \phi_c \frac{d\phi_c}{dt} \right) \tag{3.5}
\]

The valve plate opening and closing to the ports is modelled with the steady state equation for a turbulence restrictor according to equation (3.6).

\[
q_r = C_q A \sqrt{\frac{2}{\rho} \Delta p_r} \tag{3.6}
\]

\(\Delta p_r\) is the pressure drop over the valve. \(C_q\) is the flow coefficient and is known to be a function of the Reynolds number and the area difference between the orifice and the pipe. The value is difficult to estimate in the pump environment and the standard value for turbulent hole orifice of 0.60, [84], is therefore chosen. The equation is not quite valid for non-linear dynamic flow [64, 110], but in most cases the equation is justified. The steady state equation is extended in paper [VI] to be valid even for unsteady flow.

### 3.2 Unsteady flow through a valve plate restrictor

The simple steady state equation is well understood and valid for most cases. Unsteady flow, which is created for example in fluid power motors and pumps can cause some additional behaviour, e.g. [37, 64, 106, 111]. The dynamic behaviour of a fluid power component is strongly dependent on the hydraulic inductance and hence the inertia of the fluid in the passageway [62].

The steady state restrictor equation can be extended with a dynamic internal mass term, as illustrated in figure 3.3. The extension is presented in paper [VI] applied on a valve plate opening while in this section a more general derivation of a dynamic restrictor is shown. The internal mass is also called hydraulic
Steady state orifice

\[ \Delta p_1 \]

Internal mass

\[ \Delta p_m \]

Figure 3.3 Illustration of the steady state orifice and the internal mass model, including the different pressure indexes.

inductance. It can be seen as a jet stream and can be modelled as a mass-spring-damper system applied to the hydraulic system. The equation neglects the effects of fluid compressibility.

The balance forces of the internal mass can be expressed as equation (3.7).

\[
\frac{d}{dt} (m_j \dot{v}) = A \Delta p_m = A (p_2 - p_3) \tag{3.7}
\]

The parameters are shown in figures 3.4 and 3.3.

Figure 3.4 Parameter definition for calculation of the dynamic response.

The length of the internal mass, \( l_c \), is expressed as:

\[
l_c = l_r + \gamma d_h \tag{3.8}
\]

The orifice length in a valve plate restrictor can be assumed to be \( l_r \approx 0 \). In [96], the circular restrictor should be lengthened by \( \frac{\pi}{4} < \gamma < \frac{\pi}{8} \) due to jet stream effects. These values are for a single expansion or contraction while in the valve plate restrictor; the end correction is for a double-ended restrictor and hence twice as big. The value varies due to the length of the restrictor and the value is developed from an infinite pipe wall diameter. The flow through the restrictor is assumed to be constant, thus \( v \propto 1/A \).

Equation (3.7) is derived in equation (3.9).

\[
\frac{1}{A} (m_j \ddot{v} + \dot{m}_j v) = (p_2 - p_3) \tag{3.9}
\]
In equation (3.10), the pressure $p_2$ is eliminated by using equation (3.9), together with the restrictor’s steady state equation, equation (3.6).

$$\frac{1}{A} (m_j \dot{v} + \dot{m}_j v) + \frac{\rho}{2} \left( \frac{v}{C_q} \right)^2 = \Delta p_{tot}$$  \hspace{1cm} (3.10)

$q$ is expressed as equation (3.11) and the derivation of $q$ becomes as equation (3.12).

$$q = Av$$  \hspace{1cm} (3.11)

$$\dot{q} = \dot{A} v + A \dot{v}$$  \hspace{1cm} (3.12)

If equations (3.11) and (3.12) are inserted in equation (3.10), the dynamic restrictor equation can be formulated according to equation (3.13).

$$\Delta p_{tot} = \frac{m_j}{A^2} \dot{q} + \frac{1}{A^2} \left( \dot{m}_j - m_j \frac{\dot{A}}{A} \right) q + \frac{\rho}{2} \left( \frac{q}{C_q A} \right)^2$$  \hspace{1cm} (3.13)

By inserting inductance $L$ and resistance $R$, the equation can be reformulated to equation (3.14).

$$\Delta p_{tot} = L \frac{dq}{dt} + Rq + \frac{q^2 \rho}{2C_q^2 A^2}$$  \hspace{1cm} (3.14)

where

$$L = \frac{m_j}{A^2}$$  \hspace{1cm} (3.15)

$$m_j = \rho l_e A$$  \hspace{1cm} (3.16)

$$R = \frac{\dot{m}_j - m_j \dot{A}}{A^2}$$  \hspace{1cm} (3.17)

$l_e$ and the critical parameter $\gamma$ are not easy to determine. In [63] the effective length is numerically determined for zero mean flow in different orifices. According to calculations made in [63], the factor $\gamma$ should decrease by the level of contraction. However, due to the complexity of deciding the effective length, $\gamma$ can assume a constant value. Equation (3.14) is always valid, but gives only a clear visible effect at unsteady heavy flow pulsations.

The inertia of fluid in the passageways strongly influences the dynamic behaviour of the component and system and is therefore important to consider in dynamic simulations. The inertia can be considered to be equivalent to the electric inductance where the pulsating pressure is equivalent to pulsating voltage and flow ripple equivalent to pulsating current. The hydraulic inductance is not easy to quantify, neither experimentally nor theoretically, but some qualified attempts to find the impedance\footnote{The impedance is strongly affected by the inertia and the hydraulic inductance.} of different components by experiments.
have been performed in [26, 59]. The authors found that the impedance can be represented by resistive, capacitive, and inductive terms, but the terms are difficult to verify. With the increase in CFD simulations the inductance in valves can be better quantified e.g. [62, 74].

In figure 3.5, the frequency of the oscillations are shown at different pressures and rotational speeds. The machine is geometrically scaled to different displacement, i.e. all pump sizes have the same ratio between valve plate openings and piston radius etc.. Frequency will change during the rotation mainly due to change of the valve plate opening area and this is shown in figure 3.6(a). The location of the oscillation number in figure 3.6(a) is shown in figure 3.6(b).

![Figure 3.5](image)

**Figure 3.5** The frequency of oscillation with different discharge pressures and displacements. The dark grey and light grey surfaces show different rotational speeds, 1,000 rpm and 2,000 rpm respectively.

![Figure 3.6](image)

**3.6(a)** Oscillation frequency  **3.6(b)** Oscillation number

**Figure 3.6** The change in frequency of the oscillation when the cylinder size and restrictor opening area change. The oscillation number is illustrated in the right figure. The operational condition is 40 cm$^3$/rev, 2000 rpm, and 200 bar.
3.3 3D CFD simulation model

The 3D CFD-model is created in the commercial program PumpLinx from Simerics, Inc [18]. The code is based on Navier Stokes formulation of a controlled volume. The simulation program is mainly built for positive displacement pumps and motors. The program has been tested for different types of pumps/motors, mainly from a cavitation point of view, e.g. [39, 52, 82].

PumpLinx is developed with full Object Oriented Design. The model setup, grid generator, solver, and post-processor are integrated in one GUI. A three dimensional CAD representation is created and then imported in PumpLinx. The user defines the different features of the pump, i.e. piston, outlet, inlet, etc. while the program creates the grid.

The program uses body-fitted binary tree meshing which belongs to the family of unstructured body-fitted Cartesian grids. The majority of the grids are cubic but when approaching higher geometric resolution areas, an overlaid cubic grid is refined by a factor of two. The grid is cut to follow the boundary of the fluid surfaces. One benefit with the used grid is that it is possible to automate the grid set-up.

All CFD programs are based on the fundamental conservation of mass, momentum, and energy. A pump model also needs to consider turbulent flow and cavitation behaviour, which imply useful information about how the flow behaves in the cylinders and pipe lines. The program uses the standard $k-\varepsilon$ turbulent model formulated in [75]. The cavitation model formulated in [5, 102] is used and describes the cavitation vapour distribution, in section 3.4 further discussion is presented. The state-of-the art solver is used.

One additional problem with positive displacement CFD-models are the rotating parts which create relative motion of the grid structure. This has been a big challenge for the CFD programmers. The program uses an implicit method where the user defines the shared boundaries and the program creates so called mismatched grid interfaces. This interface connects the two sides at every time step. During the simulation process, the face is treated as a normal connection between grids in the same volume. A more complete explanation and technical details of the program can be found in [18].

3.3.1 The simulation set-up

The simulation set-up in PumpLinx for source flow simulations is illustrated in figure 3.7. This is comparable with the 1D model and rarely no interaction from an external system and loads. The valve plate’s thickness is 1 mm. The line dynamics are purely inside the cylinders.

Setting up the model takes about 45 minutes when changes have been made, such as valve plate parameters. The model has to run at least three rotations to converge and after that do the simulation for some periods to get satisfactory results, in total roughly 6 hours. The model above was run with 400,000 cells.
3.4 Air release and cavitation

Air release and cavitation affects the noise level in a fluid power machine. In general these are phenomena which are non-linear and includes many parameters which are hard to estimate, even with a qualified guess.

Air release occurs immediately the pressure falls below the fluid power system’s saturation pressure level. The air appears as bubbles and when the pressure increases the air is dissolved into the air. The saturation pressure level is the pressure where equilibrium is reached depending on pressure and temperature. At this point all air is dissolved into the oil. The saturation pressure level is not only a fluid parameter but is dependent on the system configuration, see [23]. For most systems, the saturation pressure level is equal to the atmospheric pressure at 0.1 MPa due to the open air reservoir. At this pressure level the oil can dissolve as much as 10% volume fraction of air but the air has negligible effects as long as the air is dissolved in the oil.

The oil starts to boil if the pressure falls below the vapour pressure level and vapour filled bubbles appear in the oil. This phenomenon is called cavitation and can cause severe damage. The phenomenon appears normally most often in valves but is an important occurrence in hydraulic machines. The vapour filled bubbles are either symmetrically collapsed far from any boundaries and create a shock-wave in the fluid or asymmetrically collapsed at or close to solid boundaries and create micro-jets, [67]. Neither of these phenomena alone can cause severe cavitation damage but a cloud of air bubbles which are triggered by each other can cause severe damage and noise.

The released air and vapour bubbles change the oil properties, especially the bulk modulus, and hence the dynamic behaviour of the simulated pump and motor. The reduced bulk modulus will increase the compressible flow pulsations and also decrease the volumetric efficiency of the machine. Air release and cavitation can appear in fluid power pumps and motors under...
certain conditions. The air release can be associated to the design of the valve plate; the opening to ports is delayed and the cylinder has no oil to imbibe or there is a big pressure difference when the cylinders connect to the ports and a jet-flow with high dynamic pressure and consequently low static pressure, and hence air, is released. The prospective air dissolving period and potential cavitation during the pressurisation can be more or less severe depending on where the air and vapour release appears. The most common cause of air release and reduced efficiency of the hydraulic system is caused by the reduced inlet pressure due to rotational speed where no boost pressure at the suction port is used. Most of the research work is done to increase the self priming rotational speed of the hydraulic pump through investigation and redesign of the inlet channel design. CFD calculation in a 3D program is very useful for such investigations. In [34], a pre-expansion filter volume to increase the self-priming speed is investigated with good results.

The effective bulk modulus in [122] is calculated according to equation (3.18) for isothermal cases. The equation can be easily changed for adiabatic processes with temperature variations, equation (3.19).

\[
\beta_{e,i} = \frac{\beta_{oil}}{1 + \frac{mRT}{p}\left(\frac{2\mu}{p} - 1\right)} \quad (3.18)
\]

\[
\beta_{e,a} = \frac{\beta_{oil}}{1 + \frac{x_0 \beta_{oil}}{\kappa p^{1-x_0} \left(\frac{\mu}{\rho}\right)^{\kappa}}} \quad (3.19)
\]

\(x_0\) is the volume fraction of free air in the oil at reference pressure \(p_0\). The efficient bulk modulus in HOPSAN of the oil and air bubble mixture is modelled according to this equation. The pressure oscillations can be assumed to be an adiabatic process and at normal hydraulic pressure levels, a realistic Polytropic exponent is about 1.8. The model of the free air content is valid for \(x_0 \leq 0.1\), [92]. In [65], the model was tested by simply compression measurements with good agreement.

The majority of the simulations and measurements in this project are made with boost pressures where no major air bubble release should appear. However, the free air content is usually between 0.8% up to 1.2% depending on the inlet pressure. The air content increases considerably in the high pressure port when the inlet pressure is decreased for increased rotation speed. Also, different valve plate designs change the bulk modulus and hence the free air content in the high pressure port. This is shown in paper [V].

Cavitation and the vapour bubbles are not considered in the 1D simulation and because of the boost pressures the creation of the vapour should be negligible. In paper [V] the pressure is expected to be bigger than the vapour pressure and hence the cavitation should be limited.

The cavitation model in the CFD program is formulated in [102] and de-
scribes the cavitation vapour distribution.

\[
R_c = C_e \frac{\sqrt{K}}{\rho_l \rho_v} \left[ \frac{2}{3} \frac{p_v - p}{\rho_v} \right]^{\frac{1}{2}} (1 - f_v - f_g) \tag{3.20}
\]

\[
R_c = C_c \frac{\sqrt{K}}{\rho_v \rho_l} \left[ \frac{2}{3} \frac{p - p_v}{\rho_l} \right]^{\frac{1}{2}} f_v \tag{3.21}
\]

\( C_e \) and \( C_c \) are empirical constants. This formulation is implemented in PumpLinx with good agreement to measurements. However, there are many unknown parameters which are not elementary to estimate.

Figure 3.8 shows simulated outlet and inlet flow pulsations with the 1D model and CFD model with different inlet pressures. The discharge flow ripple is affected due to the bigger compressible flow pulsation when the oil in the cylinder has to be more compressed to reach the discharge pressure level. However, both the 1D model and the CFD model have similar behaviour. On the contrary, the inlet flow pulsation is not correctly simulated in the 1D model, proving that the CFD cavitation model is more accurate. Cavitation phenomena and the behaviour in the inlet channel may be better studied in a full 3D CFD simulation model where more cavities are simulated.
Figure 3.8 Internal mass simulation with 1D model compared with CFD-model, dashed line and solid line respectively. The rotational speed is 2000 rpm and outlet pressure is 200 bar with inlet pressure 5 bar and 30 bar respectively. A zero-lapped valve plate is simulated.
4 Optimisation

Computers’ calculation capabilities are increasing and hence the ability to make numerical optimisations. By using an optimisation algorithm, a systematic way is applied to find the “best” fitted solution for the criteria which have been stated in the formulation of the optimisation problem. An optimisation problem includes at least one objective function which includes the goal of the optimisation. The variables which are used to find the optimal solution are called design variables. In an engineering problem the searchable area is most often limited: the limitations are called constraints. The goal for the “best” solution when designing a product can be to maximise the efficiency or minimise the price while the constraints can be manufacturing issues. However, most often both the price and the efficiency are important for a good product, hence the optimisation is rarely a single-objective optimisation problem.

4.1 Multi-objective optimisation

There is generally more than one interesting characteristics to optimise in most engineering problems and a multi-objective optimisation has to be formulated, equation (4.1).

\[
\begin{align*}
\min F(x) &= (f_1(x), f_2(x), \ldots, f_n(x))^T \\
\text{s.t. } x &\in S \\
x &= (x_1, x_2, \ldots, x_m)^T
\end{align*}
\]

where \( f_1(x), f_2(x), \ldots, f_n \) are \( n \) objective functions, \( x_1, x_2, \ldots, x_m \) are \( m \) optimisation parameters or design variables, and \( S \in R^m \) is the solution or parameter space which is limited by boundaries or constraints to avoid undesired behaviour. In a general design problem, \( F \) can be non-linear and \( S \) can also be non-linear and contain both continuous and discrete variables.
The utopian solution of the multi-objectives optimisation problem is to simultaneously minimize all the individual objective functions. However, the solution is rarely feasible and also the objective functions are rarely independent of each other. To determine the optimal solution, additional criteria have to be formulated and an aid in this is the concept of Pareto optimality. Along the Pareto optimal front in figure 4.1(a), there are non-dominated solutions and the front shows the trade-off between the different objective functions. There is no better solution which is simultaneously better in both objectives, hence there is a rational reason to choose a point on the Pareto optimal front, e.g. $P_1$, $P_3$ or $P_4$, rather than $P_2$ in figure 4.1(a). However, there is no rational reason to choose $P_1$ in favour of $P_3$ or $P_4$.

![Diagram of Pareto optimal fronts](image)

**Figure 4.1 Illustration of Pareto optimal fronts.**

In figure 4.1, a weak and a strong Pareto optimal front are shown. The trade-off between the two competitive objective functions is clearly feasible in figure 4.1(a), if $f_2$ reduces $f_1$ is significantly increased. By contrast, figure 4.1(b) shows a weak Pareto front. It is only close to $f_{1,\text{min}}$ that the competitive objective function $f_2$ increases rapidly. Point $P_3$ is almost as good as $P_4$ if $f_2$ is considered but with a huge improvement in $f_1$, hence there is no obvious reason to choose point $P_4$ rather than point $P_3$.

Optimisation methods can be divided into derivative and non-derivative methods. A non-derivative method is generally more suited for design problems and simulation based optimisation due to the lack of derivatives for the objective functions. Furthermore, they are more robust compared to the derivative methods in locating the global optimum, [31] and can be applied to a wider set of optimisation problems. A disadvantage is that it can not be proven that the actual optimum is found, but by applying different start values the global optimum can be considered to be found. Also, the non-derivative methods are in general computationally heavy. In this thesis the non-derivative method, the Complex method, is used [14, 33, 68]. For a comparison of different non-derivative methods see [12, 51]. The complex method has earlier been applied...
to different research areas such as fluid power systems, aerospace engineering, and physics. The method has previously been used in fluid power systems design optimisations, [4, 54].

The Complex method is a single objective optimisation method. Multi-objective optimisation problems, like equation (4.2) have to be reformulated using for example a weighted sum with weighting factors $\lambda$ according to equation (4.3). The constraints are added to the objective function using penalty functions and discrimination factor $\vartheta$.

$$
\min F(\Psi) = (f_1(\Psi), f_2(\Psi), \ldots, f_m(\Psi))^T
$$

$$
\Psi^l_i \leq \Psi_i \leq \Psi^u_i, \quad i = 1, 2, \ldots, m
$$

$$
g_j(\Psi) \leq 0, \quad j = 1, 2, \ldots, n
$$

$$
\min F(\Psi) = \sum_{i=1}^{m} \lambda_i f_i(\Psi) + \sum_{j=1}^{n} \vartheta_j g_j(\Psi)
$$

Depending of the reason for the formulated restriction, $G_j$ can be formulated as soft or hard constraints, equation (4.4) and (4.5) respectively.

$$
G_j(\Psi) = \max [0, g_j(\Psi)]
$$

$$
G_j(\Psi) = \{1, 0\}
$$

Soft constraints express some certain preferences, continuous variables and hard constraints should always be satisfied, boolean variables.

**4.2 Optimisation in respect of noise reduction**

As stated earlier in the thesis, noise in hydraulic pumps and motor is caused by many different mechanics and the noise minimisation problem is therefore a typical multi-objective optimisation problem. It is preferable to use an optimisation algorithm in favour of manually, albeit systematically, finding the optimum solution. The optimisation algorithm should be better to find the global optimum and search in a wider design space with no personal opinions. A manual optimisation is limited to the designer’s ability to understand couplings between different objectives and design variables. In [55] Generic Algorithm is used to minimise flow pulsation, axial piston force and swash plate moment in hydraulic in-line pump. [99, 100] use a model similar to a Generic Algorithm to find the optimal solution of several different objectives.
The constraints for pump and motor optimisation depend on the chosen objective function and the design feature but some general constraints can be stated which are used throughout the thesis.

- **Cavitation** - The minimum pressure level is limited to prevent cavitation. In the objective function, cavitation is treated as a soft constraint, i.e. the pressure may be below the pre-defined minimum pressure level under some operating conditions, but there will never be any severe low pressures.

- **Pressure overshoot** - The pressure is not allowed to be above a certain pre-defined level and is treated as a soft constraint, i.e. the pressure may be above the pre-defined maximum pressure level under some operating condition, the objective function value will be increasingly penalised with the pressure overshoot.

- **Cross-flow**\(^1\) - Flow between the ports is not allowed and thus the discrimination factor is of boolean variable and the constraint is hard. The cross-flow will increase the volumetric losses in the pump and due to the fact that efficiency is not treated in the work, all cross-flow through optimisation of the different features is eliminated. The cross-flow will likely reduce both the efficiency and flow pulsations to some extend.

### 4.2.1 Objective function

There are several different objectives which could be considered for noise reduction in fluid power pumps and motors e.g. axial piston force, bending moments, inlet and outlet flow pulsation, derivative of axial piston force and pressures, not to mention emitted audible noise. The list of presumptive characteristics to use for noise minimisation can be long and all generate noise in the origin pump/motor unit. In the study in [99] only time domain variables are used and these are flow pulsation in high and low pressure ports, one bending moment, and the volumetric losses due to the allowance of cross-flow, which reduces the volumetric efficiency of the pump. The most used objective function when the flow pulsation reduction is considered and hence the the created noise is obviously the amplitude in the time domain of the flow pulsation in high and low pressure ports.

The main question is which more general and which of the characteristics are cause the most noise. To make it more complex and difficult to interpret, the two questions are most probably system dependent. However, a pump motor unit is most often designed without knowledge of a particular system which makes it suitable to investigate the optimal objective function for the pump and motor unit itself and try to include different kinds of systems in which

---

\(^1\)Cross-flow implies a flow between the high pressure port and low pressure port at the area where the cylinder disconnects and connects to respective port.
the machine will probably be mounted. All investigations during the project have neglected the system dependency but the solution of the pump and motor designs should be suitable in different kinds of systems.

Probably the most obvious objective function when minimising the fluid borne noise created inside the machine is to minimise the peak-to-peak value of the flow pulsation in time domain, equation (4.6).

\[
\min f(\Psi) = \lambda (\max (q_h (t)) - \min (q_h (t))) + (1 - \lambda) (\max (q_l (t)) - \min (q_l (t)))
\] (4.6)

The Pareto optimal front for the high and low pressure port flow pulsation is weak, i.e. both the objectives have to be considered to minimise the fluid borne noise but the value of the discrimination factor is of less importance.

The perceived noise is different at different frequencies [11]: it may therefore be preferable to formulate the objective function in the frequency domain. In [61], the suggestion for rating noise emission is the square sum of each harmonic of the amplitude of the frequency spectrum, equation (4.7), which may be a misleading quantity due to the human interpretation of high frequency noise being bigger compared to low frequency noise.

\[
\min f(\Psi) = \sum_{k=1}^{n} (Y_k(j\omega))^2
\] (4.7)

In addition, shown in [95], the transformation from axial piston force and bending moment into audible noise at the pump housing are higher at 1000 Hz compared at lower frequencies which restate equation (4.7) if these objectives are chosen. It is also shown in [95] and section 6 that the total axial piston force has a larger impact on the noise level at the pump housing compared to the bending moment.

An early reference [105] suggests calculating a single figure of merit based on an A-weighted filter [11] which takes the human interpretation of noise into consideration. This is done in papers [III] and [IV] for the audible noise minimisation. For paper [III] when the amount of harmonic frequencies with energy content are kept constant, the A-weighted filter value is probably a good value to compare different optimised designs. In paper [IV], when the pump is made non-uniform, the amount of frequencies containing energy increases. The harmonic frequencies with energy content are close together and different phenomena appear, e.g. masking effects. These phenomena and also others can probably change the interpretation of the noise.

The resulting axial piston force should be as smooth as possible in order to minimise the overall noise generated in a hydraulic pump motor unit. This means that the cylinder pressure derivative \( \frac{dp}{dt} \) should be kept small, equation (4.8). The cylinder pressure rate was suggested in [43] and [53]. The pressure rate is ranked as the best overall objective function in paper [II] when
both the high pressure and low pressure ports’ pressure rate is considered.

\[
\min f(\Psi) = \max \left( \frac{dp}{dt}(t) \right) - \min \left( \frac{dp}{dt}(t) \right)
\]  

(4.8)

### 4.2.2 Audible noise as an objective

The optimal objective function to minimise the audible noise at the pump housing should be to use that particular characteristic as an objective function, which is done in paper [III]. The transfer function method which transform forces inside the machine into audible noise, is discussed later in chapter 5. Below, the audible noise as an objective is discussed. The sound pressure level is most often expressed in decibels and can be calculated as

\[
L_{pn} = 10 \cdot \log \frac{p^2}{p_{ref}^2}
\]  

(4.9)

where \( p_{ref} \) corresponds to the minimum audible sound pressure at 1000 Hz and is \( 2 \cdot 10^{-5} \) Pa.

Octave bands are often used to get a cleaner view of the frequency contents. The centre frequencies of each band are standardised and are usually 31.5 Hz, 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1 kHz, 2 kHz, 4 kHz, 8 kHz, and 16 kHz. The harmonic frequencies from the simulation and measurements are collected to each band with equation (4.10). The collection of frequencies from simulation of the audible noise emitted from a pump is shown in figure 4.2.

\[
L_p = 10 \cdot \log \sum_{n=1}^{N} 10 \frac{L_{pn}}{10}
\]  

(4.10)

![Figure 4.2](image_url)

**Figure 4.2** A frequency spectra from sound level at 1700 rpm, solid line with the corresponding Octave band, dashed line.

To adjust sound measurements corresponding to loudness as perceived by the average human ear, a weighting filter is used. The weighting filter emphasises
frequencies between 1-6 kHz where the human ear is most sensitive while lower and higher frequencies are attenuated where the human ear is less sensitive.

Different weighting filters are used to adjust the measured sound pressure level to loudness as perceived by the average human ear. The A-, B-, and C-weighting functions are developed from 40-, 70-, respective 90-phon Fletcher-Munson curves and should be used for different noise levels; however, only A- and C-weight filters are commonly used. C-filter is used for very high noise levels while A-filter is generally used and is preferable for sound pressures at the pump housing. Figure 4.3 shows the three most common weighting filters.

![Figure 4.3](image_url)

**Figure 4.3** A-, B-, and C-weighted filters to adjust the sound pressure level to loudness as perceived by the average human ear. A-filter is the most commonly used.

The sound pressure level with A-weighting is calculated as

\[
L_A = 10 \cdot \log \sum_{n=1}^{N} 10^{\frac{L_p + \Delta A_n}{10}}
\]  

(4.11)

where \(\Delta A_n\) refers to the solid line in figure 4.3.

Finally, the audible noise is minimised as equation (4.12).

\[
\min f(\Psi) = L_A
\]  

(4.12)

Paper [III] considers the audible noise at the pump housing, i.e. structure borne noise at the pump shell, as an objective function. The paper shows that the cylinder pressure derivative \(dp/dt\) is good for minimising audible noise at the pump housing. The same paper shows that the square sum of each harmonic of the amplitude spectrum for the derivative of the total axial piston force, equation (4.7), is a good minimiser of the audible noise at the pump housing. Audible noise optimisation increases the complexity of the simulation based optimisation and the simplest second hand choice is the pressure rate in the time domain or the total axial piston force in the frequency domain.

Despite the overall general objective functions, it is important to consider both the fluid borne noise and structure borne noise, i.e. fluid pulsations and
audible noise respectively at the pump housing, because of the different system dependency of the characteristics. The structure vibration is most often decreased if the flow pulsation is minimised and also vice versa.
5

Source flow measurement

The main purpose of the source flow measurement is to validate the simulation model. However, it is also important to be able to rate the noise created in the pump and motor unit. Due to the fluid borne noise’s ability to spread in the external system, the flow pulsations are essential when rating the noise created in the pump/motor unit. The flow pulsation is supposed to have bigger impact of the external system compared to the structure born noise at the pump housing. It is therefore extremely important to consider the source pulsation because the flow pulsations created at the pump can travel through the pipe line system and excite noise in many different places. The flow pulsations can also be enhanced by other components.

It is important to eliminate the external system’s influence of the experimental test set-up where the test object is mounted to be able to validate purely the motor/pump unit’s excitation of noise, regardless of whether structure borne noise or fluid borne noise is being considered. The external system is important for an overall noise excitation from the hydraulic system; however, when designing a motor/pump unit, it is very rare to know exactly where the machine will be placed and a good start will be to have a quiet motor/pump unit.

There is no flow meter which can resolve the desired frequencies. Pressure pulsations therefore have to be measured and thereafter transformed in flow pulsations. The pressure ripple is generally accepted as a rating value of fluid borne noise. However, the pressure ripple is not caused by the pump/motor unit itself but an interaction with the external system. Hence, it is not possible to determine how much of the pressure ripple originates from the pump itself. Also, the simulation model of the pump created in HOPSAN can not be validated by the pressure ripple measurements; the source flow therefore has to be measured. Source flow is understood to be created at the valve plate and
is independent of the dynamic load. To get a complete description of the fluid borne noise created in the pump, both source flow and source impedance are needed, see section 5.1 for further explanation.

There are different methods to transform pressure measurement into flow pulsations and further into source flow and source impedance quantities. One example is the hydraulic trombone method developed at the University of Bath [21]. The method uses different pipe lengths. The source characteristics can be separated with a mathematical algorithm, but the method is rather time-consuming. Another method is the secondary source method [24, 25]. This method has been adopted as the international standard for high-precision measurements of fluid-borne noise from hydraulic pumps. The method uses a secondary source which produces a broadband spectrum in the region of the measurement unit that is of interest but with harmonics different to the test unit.

In the thesis, the flow pulsation has been measured with the two-microphone method to validate the simulation model. The anechoic termination method is an example of a measurement method which does not separate the source impedance and source flow into two quantities. The method gives a good value of the flow pulsation provided that the outlet channel’s source impedance is large. The method is very simple and is therefore used in combination with the transfer methodology measurements in the thesis. A novel method called the source admittance method is also investigated. An admittance matrix is used to model the pump’s outlet channel, which explains the name of the method. In the following sections, the source flow measurement techniques which are used in this thesis are examined.

5.1 Source characteristics

The flow pulsation which is created at the valve plate is called source flow and is completely system independent. The source flow exclude also the pump’s outlet channel. This means that source flow is constant for any dynamic load with the same stationary pressure. The simulation model built in HOPSAN calculates the source flow; to validate the simulation model the source flow therefore needs to be measured. To describe the motor/pump unit noise contribution the impedance is also needed, which is the geometrical compartment in the pump channel, i.e. between the valve plate and the pump flange. The pump’s outlet channel is illustrated in figure 5.1. \( Z_s \) is the source impedance and \( Q_s \) is the source flow. With the obvious analogy fluid dynamics and electricity, Norton equivalents can therefore be used to obtain the relationship between pressure and flow. Figure 5.1 can be formulated mathematically as equation (5.1) and (5.2).

\[
Q_s = Q_1 + \frac{P_1}{Z_s} \quad (5.1)
\]
Source flow measurement

Figure 5.1 The impedance representation on which equation (5.1) and equation (5.2) is based on.

\[ \frac{1}{Z_1} = Q_s P_1 - \frac{1}{Z_s} \]  

(5.2)

5.2 Model of wave’s propagation in pipeline

To model the unsteady flow between two points through a rigid pipeline a four-pole matrix is used, equation (5.3). The model is a two dimensional viscous compressible model which includes frequency dependent friction losses. The four-pole matrix is a linear system of equations relating pressure and flow at each side of a pipe. The equation is in frequency domain and containing complex elements. Figure 5.2 shows the flow direction and indexing definition for equation (5.3).

\[
\begin{pmatrix} Q_2 \\ P_2 \end{pmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{pmatrix} Q_1 \\ P_1 \end{pmatrix}
\]  

(5.3)

where

\[
A = D = \cosh(T s \sqrt{N})
\]

\[
C = -\frac{1}{Z_c \sqrt{N}} \sinh(T s \sqrt{N})
\]

\[
D = -Z_c \sqrt{N} \sinh(T s \sqrt{N})
\]

(5.4)
On Fluid Power Pump and Motor Design

\[
T = \frac{L}{a} \quad a = \sqrt{\frac{p_0}{\rho}} \quad Z_c = \frac{\rho a}{\pi r^2} \quad s = i \omega
\]

(5.5)

The important limitations of the four-pole matrix are that the flow is assumed to be laminar and the pipe is non-elastic. Consequently, tangential flow is neglected and the radial speed is also neglected because pressure and density change far more in the axial direction than in the radial direction. The four-pole matrix and the model’s limitations are further explained in [109]. There are other ways to solve the pipeline model in a stricter way and with fewer assumptions but the simplifications are justified, according to [109]. When considering source flow measurement, the inner diameter of the measurement pipe should be equal to the pump outlet channel to avoid resonance in the pump’s outlet channel caused by reflections at this point.

The four-pole matrix is used to transform the measured pressure pulsation along the measurement pipeline to flow pulsation.

5.3 Anechoic termination method

One of the oldest flow pulsation measurement methods is the anechoic termination method [73]. The anechoic termination, a volume and orifice combination, creates a reflection-free system. Given that the viscous friction can be ignored, the pressure wave at any position along the pipeline is identical except for the time shifts due to the wave’s propagation time. The method is very simple and is probably the most used method together with the high impedance pipe test method. This method realised through choosing the pump flange impedance \(Z_1\) much larger than the source impedance. Thus, equation (5.2) becomes as shown in equation (5.6) and the source flow is proportional to the measured pressure at the pump flange.

\[
P_1 \approx Q_s Z_2
\]

(5.6)

The high impedance pipe test method is adapted to the international standard for rating fluid borne noise generated in pumps [48]. Neither the anechoic termination method nor the high impedance pipe test method gives the source impedance of the outlet channel and thus does not give a complete description of the pump fluid borne noise contribution.

Figure 5.3 shows a simplified illustration of how a measurement set-up can be arranged to perform anechoic termination measurement at different stationary pressure levels. The orifice \(k_{c2}\) is used to set the stationary pressure independently of orifice \(k_{c1}\) which is used to tune the set-up to anechoic condition. The linearised restriction coefficient for \(k_{c1}\) placed at the end of the measurement...
pipe can be formulated as:

\[ R_{e1} = \frac{1}{k_{c1}} = \frac{P_2}{Q_2} = \frac{2\Delta p}{q_2} \]  

(5.7)

where \( \Delta p \) is the pressure drop over the orifice and \( q_2 \) is the flow through the same orifice. To produce anechoic condition \( R_{e1} \) is tuned so that \( Z_c \) equals \( R_{e1} \).

Equation (5.8) is formulated by dividing the two equations in matrix (5.3) with each other and ignoring the frequency dependent friction, i.e. set \( \sqrt{N} = 1 \).

\[
\frac{Q_1}{P_1} = \frac{1}{Z_c} \left( Z_c \cosh(T_s) + \frac{P_2}{Q_2} \sinh(T_s) \right) / \left( \frac{P_2}{Q_2} \cosh(T_s) + Z_c \sinh(T_s) \right)
\]  

(5.8)

At anechoic condition, \( Z_c = R_{e1} \), equation (5.8) becomes:

\[
Q_1 = \frac{P_1}{Z_c}
\]  

(5.9)

Hence, the dynamic pressure measured at position 1 is proportional to the dynamic flow at the same place as well as any other positions along the pipeline. The method measures neither the source impedance nor the source impedance. However, if the source impedance is sufficiently large, the measured flow at the pump flange and the source flow can be considered to be the same according to equation (5.1). This is also shown in figure 5.4 where simulated source flow is compared to the flow at the pump flange measured at anechoic condition. The simplicity of measurement at anechoic condition makes it suitable to rate flow pulsation for different pump design with the method.

### 5.4 Two-microphone method

The two-microphone method originates from the acoustics area [16]. The method was developed at Linköping University for source flow measurements [112]. The benefit with the two-microphone method is that the technique separates the source flow and source impedance as two different quantities and hence can be used for verification of simulation models. Also, compared to other similar methods such as the secondary source method the main test pump is used...
to introduce a broadband frequency spectrum instead of a separate unit. The disadvantage of these methods is that a mathematical model of the pump's outlet channel is needed.

The two microphone method uses the four-pole matrix presented in equation (5.3) and hence the pressure has to be measured at two places along the measurement pipe. If the two equations in the four-pole matrix are combined and $Q_2$ is eliminated, equation (5.10) is obtained.

$$Z_1 = \frac{P_1}{Q_1} = \frac{Z_c \sqrt{N} \sinh(Ts\sqrt{N})}{\cosh(Ts\sqrt{N}) - \frac{P_2}{P_1}}$$  \hspace{1cm} (5.10)

Figure 5.4 Source flow measurements at anechoic condition with a bent-axis pump with large source impedance at two different rotational speeds. Stars and bars show measurement and simulation respectively.

The measurement set-up, shown in figure 5.5, is very similar to the anechoic termination set-up with one additional pressure transducer. Two measurements are performed to obtain two different dynamic load cases at the same stationary pressure level. This implies that the pressures are measured at two different reflections of the pressure wave. The reflections create two different pressure profiles and two linear independent equations of the source impedance can be formulated with equation (5.2). The load cases, indexed by $L_1$ respective $L_2$, is put into practice through changing the pipeline end impedance $Z_T$. This is done by varying the orifice's opening area.

Figure 5.5 Measurement set-up for the two-microphone method.

To obtain sufficiently different loads the end impedance is recommended to
be $Z_{TL1} = 2Z_c$ and $Z_{TL2} = 0.5Z_c$ \[92\].

By combining equation (5.10) and equation (5.2) with the two dynamic load cases, the source impedance and source flow can be formulated as different quantities as shown in equations (5.11) and (5.12).

$$Z_s = \frac{P_{1L2} - P_{1L1}}{Z_{1L1} - Z_{1L2}}$$ (5.11)

$$Q_s = \frac{1}{P_{1L1}} - \frac{1}{P_{1L2}}$$ (5.12)

$Q_s$ and $Z_s$ can be considered to be the source flow and source impedance if the distance between the valve plate where the source flow is supposed to be created and the pump flange is negligible. In section 5.4.1, an example of a non-negligible outlet channel is shown.

In addition to the importance of the measurement pipe diameter to avoid resonance in the pump’s outlet channel there are some other issues to consider. Cancellation errors can appear at frequencies given by equation (5.13) due to standing waves in the measurement pipe.

$$\omega = \frac{i\pi}{T} \quad \text{where} \quad i = 1, 2, 3, ...$$ (5.13)

$l$ is the length between the pressure transducer at the measurement pipe whereas $a$ is the wave propagation velocity. If the harmonics of the pump, $\omega = iz\omega_p$ where $i = 1, 2, 3, ...$ (5.14) coincides with $\omega$ in equation (5.13), problems can appear with amplified amplitude of the source flow. If possible, the distance between the pressure transducers should be so placed as to avoid cancellation errors.

According to \[9, 10\], where the measurements were performed in gaseous fluid, the recommended length of the pipe in relation to the desired frequency interval, $f_{min}$ to $f_{max}$, is

$$0.05 < \frac{L}{a}$$ (5.15)

$$0.4 > \frac{L}{a}$$ (5.16)

to obtain correct measurements. However, according to \[92\] these limitations are not crucial when the measurements are performed in a liquid fluid.

The volume between the orifices has previously been used for anechoic termination and is described in \[73\]. The volume is not used for anechoic termination in this study, but is useful for separating the dynamic load from the static pressure in the system. The size of the volume is critical when used for anechoic termination but is more conservative with the proposed method. The volume
is effective for isolating the measurement pipe from higher frequencies, above \( \approx 100 \text{Hz} \), from the rest of the system.

In figure 5.6, measurements are shown with the two-microphone method where the source impedance is modelled as in section 5.4.1.

![Figure 5.6](image)

Figure 5.6 Source flow measurements with the two-microphone method with a bent axis pump with large source impedance at two different rotational speeds. Stars and bars show measurement and simulation respectively.

5.4.1 Modelling a non-negligible outlet channel

Figure 5.7 shows how the pump outlet channel can be modelled if the distance between the pump flange and valve plate can not be ignored. The pump outlet channel can be modelled in a similar way to the previous section with the four-pole matrix. The parameters to the left in figure 5.7 are similar to figure 5.1

![Figure 5.7](image)

and therefore a Norton representation is obtained with the denotion for the outlet channel as:

\[
Q_{s0} = Q_0 + \frac{P_0}{Z_0}
\]  

(5.17)

The new source flow is denoted with \( Q_{s0} \).

\( Z_0 \) is commonly modelled with a leakage term and a capacitance as shown in equation (5.18).

\[
Z_0 = \frac{1}{\beta_0 s + k_c}
\]  

(5.18)
$k_c$ is the leakage in the pump channel where the main leakage is between pistons and cylinder walls and between valve plate and cylinder barrel. The capacitance $V_s$ includes the volume behind the valve plate and also the cylinder volumes that are connected to the discharge port. The equation can be illustrated as shown in figure 5.8.

$$
\begin{align*}
\text{Figure 5.8} & \text{ The impedance model } Z_0 \text{ that equation (5.18) is based on.}
\end{align*}
$$

The line dynamics in the pump outlet channel is modelled with the four-pole matrix equation (5.3) but for the pump’s outlet geometry, equation (5.19).

$$
\begin{align*}
\begin{pmatrix} Q_1 \\ P_1 \end{pmatrix} = \begin{bmatrix} A_p & B_p \\ C_p & D_p \end{bmatrix} \cdot \begin{pmatrix} Q_0 \\ P_0 \end{pmatrix}
\end{align*}
$$

Equation (5.17) and matrix (5.19) can be combined into

$$
\begin{align*}
\frac{1}{Z_1} = \frac{Q_{s0}}{P_1 X_1} - \frac{1}{X_2}
\end{align*}
$$

Equation (5.20) can be found by comparing equations (5.2) and (5.20).

$$
\begin{align*}
Z_{s1} = X_2 \quad \text{and} \quad Q_{s1} = \frac{Q_{s0}}{X_1}
\end{align*}
$$

Equation (5.20) can now be written as

$$
\begin{align*}
\frac{1}{Z_1} = \frac{Q_{s1}}{P_1} - \frac{1}{Z_{s1}}.
\end{align*}
$$

To be able to calculate the true source flow, $Q_{s0}$, the term $X_1$ has to be calculated. This is done by first determining the model term $X_2$ through comparison with the measured impedance $Z_{s1}$. The parameter values that can be adjusted are pipe diameter, pipe length, volume, and leakage term. When $X_2$
is determined, $X_1$ can be calculated with equation (5.21) and later $Q_{s0}$ with
equation (5.23). Source flow $Q_{s0}$ and source impedance $Z_{s1}$ in the pump do not
appear at the same spot, but individually they are the most suitable quantities
of the source characteristics. Figure 5.9 shows the measured source impedance
$Z_{s1}$ and the modelled source impedance $X_2$. The pump is an in-line axial piston
machine.

Figure 5.9 The measured source impedance $Z_{s1}$ (dots) and mathematical model
of the same impedance $X_2$ (solid line).

5.4.2 Measurement of effective bulk modulus, $\beta_e$

The two microphone method needs the effective bulk modulus, representing the
combined stiffness of the mechanical stiffness of the pipeline and the fluid. The
difficulty when determining the effective bulk modulus is to predict the amount
of free air in the oil. In [60], a method to calculate the speed of sound through
measurement of pressure at three locations along a rigid pipe is presented,
hence the method is called the three-microphone method. By using equation
(5.10), the point impedances can be calculated at all three locations. Equations
(5.25) and (5.26) show the impedance equations where index $i$ is the location
upstream of index $j$.

\[
Z_i = \frac{P_i}{Q_i} = \frac{Z_c \sqrt{N} \sinh \left( \frac{L_{i,j}s \sqrt{N}}{a} \right)}{\cosh \left( \frac{L_{i,j}s \sqrt{N}}{a} \right) - \frac{P_j}{P_i}} \quad (5.25)
\]

\[
Z_j = \frac{P_j}{Q_j} = -\frac{Z_c \sqrt{N} \sinh \left( \frac{L_{i,j}s \sqrt{N}}{a} \right)}{\cosh \left( \frac{L_{i,j}s \sqrt{N}}{a} \right) - \frac{P_j}{P_i}} \quad (5.26)
\]

By measuring the pressure at three locations the impedance can be calculated
with two different pressure combinations and equation (5.27) is obtained

\[
-P_3 \sinh \frac{L_{12}s \sqrt{N}}{a} + P_2 \sinh \frac{L_{13}s \sqrt{N}}{a} - P_1 \sinh \frac{L_{12}s \sqrt{N}}{a} + P_2 = 0 \quad (5.27)
\]
It is impossible to calculate the equation exactly to zero. Therefore, an objective function can be formulated as

\[ f = \sum_{j=1}^{n} |\epsilon_k|^2 \] (5.28)

where the error for all harmonic frequencies \( n \) is summarized. The correct speed of sound is determined by minimizing the objective function. This is done with the numerical optimization method, Complex algorithm [14]. Normally, the density can be assumed to be constant and the effective bulk modulus can therefore be calculated as

\[ \beta_e = a^2 \rho \] (5.29)

The main problem with this technique is that not all the frequency spectra of pressure are treated; truncation error can appear. This can be seen as small differences in the speed of sound between different dynamic loads at the same stationary pressure.

Figure 5.10 shows the effective bulk modulus when the inlet pressure is reduced due to the increase in rotational speed. The bulk modulus is decreased by over 50 MPa, which in turn will change the amplitude of the flow pulsation created in the outlet channel.

5.10(a) The effective bulk modulus
5.10(b) Inlet pressure

**Figure 5.10** The trend of the effective bulk modulus when the inlet pressure is decreased caused by increase in rotational speed. The bulk modulus is measured with the three-microphone method. Crosses are measurements and solid lines are quadratic equations which are adjusted to the measurements.

5.5 The source admittance method

A third measurement method which is tested and developed is the source admittance method, [53] and paper [VII, VIII]. This method uses the two-microphone
method but instead using the model of the pump outlet channel; the pressure pulsations are measured where the source flow is created, i.e. at the valve plate. The outlet channel can be difficult if not impossible to model in some cases. As an example, a time invariant source impedance as the case with pre-compression filter volumes is very difficult to model.

Figure 5.11 The admittance matrix denotation and definition of the flow direction for the pump outlet channel, equation (5.30).

The admittance matrix explains the relationship between dynamic flows and pressures in any hydraulic circuit. In equation (5.30), the admittance matrix is applied to the pump outlet channel, where \( Q_s \) is the source flow at the valve plate and \( Q_1 \) is the flow measured with the two-microphone method at the pump flange. Figure 5.11 shows the admittance matrix applied on the pump outlet channel. The elements \( h_{11} - h_{22} \) are complex quantities and due to reciprocity, \( h_{12} \) and \( h_{21} \) can be assumed equal.

\[
\begin{pmatrix} Q_s \\ Q_1 \end{pmatrix} = \begin{bmatrix} h_{11} & h_{12} \\ h_{21} & h_{22} \end{bmatrix} \begin{bmatrix} P_s \\ P_1 \end{bmatrix}
\]

Using the two different loads and equation (5.30), the source flow and the source impedance are calculated as equations (5.31) and (5.32) with the parameters \( h_{11} - h_{22} \) stated in equation (5.33).

\[
Q_s = -(h_{11}P_{sL1} - h_{12}P_{sL1})
\]

\[
Z_s = \frac{1}{h_{11} - h_{22}}
\]

\[
h_{11} = \frac{P_{1L2} - P_{1L1}}{P_{sL1} - P_{sL2}}
\]

\[
h_{12} = \frac{Q_{1L1} - h_{22}P_{1L1}}{P_{sL1}}
\]

\[
h_{22} = \frac{P_{sL1}Q_{1L2} - P_{sL2}Q_{1L1}}{P_{1L2}P_{sL1} - P_{1L1}P_{sL2}}
\]

The equations are further explained in [53].

The same pump and same operating conditions as in figure 5.4 and 5.6 are shown in figure 5.12. There is a small increase in amplitude at frequencies between 1,500 and 2,500 Hz at 2,000 rpm (figure 5.12(b)), which is not clearly seen with other measurement methods. However, in [92] similar behaviour can
be seen with the two-microphone method. In the thesis, the difference between simulation and measurement is explained as insufficient bandwidth to describe the source flow. Thus, the measurement is performed up to 3 kHz and the frequencies above are truncated in analysis. The frequency of the increased amplitude is slightly lower than the internal mass, section 3, gives rise to. However, the external system with the pipe line and volume can reduce the frequency magnitude.

The tested in-line pump shows larger differences than the bent-axis pump. Figure 5.13 shows some results from measurement with the in-line pump. Figure 5.13(a) shows source flow at low rotational speed and figure 5.13(b) at high rotational speed. The difference shown is typical for measurement with the tested pump and the measurement method. The measurement results contain at high frequency oscillation (∼2kHz) which is not present in the simulation or other measurement methods. The measurement is comparable neither with the existing simulation model implemented in HOPSAN nor the two-microphone method. It is not evident that the source admittance method measures the incorrect source flow; the simulation model is perhaps too simplified for some pumps and working conditions.

5.5.1 Reasons for the measurement behaviour

The frequency of the superimposed ripple can not be explained directly by the internal mass; thus, the frequency is too low and in addition the behaviour of the flow pulsation is not quite the same.

The similarities between the measurements and internal mass simulations is attributed to the trend of the oscillations.

- The results from measurements and simulations are more comparable at low speed and the trend is that the oscillations become worse if the speed
The oscillation seems to become worse when the flow ripple increases, especially the derivation of the flow ripple, i.e. quick flow disturbance.

- The oscillation does not appear with a good pressure relief groove added to the valve plate.
- The in-line pump's oscillation become worse than the bent-axis pump.

Divergences can also be found between the measurements and internal mass simulations.

- The frequency of the additional oscillation is lower in measurements than in simulation.
- The oscillation is larger in measurements and more unpredictable.
- The oscillation is not pressure dependent in measurement compared to internal mass simulations.

The differences between the tested pumps, a variable in-line pump and a fixed bent-axis pump, are many. The in-line pump has many more additional
moving parts than the fixed bent-axis pump. Some considered reasons for the different behaviour of the pumps was examined in [X] and a summary of the investigation is stated below.

The outlet channel

The complexity of the outlet channel is bigger for the in-line pump than the bent-axis pump. The admittance matrix assumes a linear relationship between the flows and pressures in the frequency domain. The two-microphone method has problem when the source impedance become too complex especially time variant impedance, [53]. An example of a time variant system is the pre-compression filter volume where the volume is only connected to the cylinder part of the rotation. The simple bent-axis machine has been mounted with a pre-compression filter volume. The source impedance is shown in figure 5.14.

The difficulty in creating a suitable model of the source impedance can be clearly seen, and hence to get satisfying measurement results with the two-microphone method. Neither the size nor the connection to the valve plate are optimised and the source flow is therefore not optimal. Figure 5.15 shows the source flow measured with the admittance method. To further prove the measurement method’s independence of the outlet channel shape, the channel has been extended and also with a decrease and increase in the cross-section area. None of the tested outlet channels show any dissimilarity to the simulation model. There is no reason to suspect that the channel’s appearance has any significance. More results when the outlet channel is changed can be found in paper [VIII].

Figure 5.14 The source impedance on the bent-axis pump; one original version of the outlet channel and one with a pre-compression filter volume mounted. The solid line is the mathematical model and the markers are measurements at 900, 1,600 and 2,000 rpm.

Slippers

The in-line pump’s pistons are supported by slippers which are hydro-
The source flow on the bent-axis pump; one original version of the outlet channel and one with a pre-compression filter volume mounted. The source impedances are shown in figure 5.14. Measured (squares and dots) and simulated (bars) source flow in the frequency domain. The operating condition is 1,600 rpm and 200 bar.

Figure 5.15

The slippers can loosen from the swash-plate and cause small displacements of the axial motion of the pistons and flow disturbances. The oscillation caused by the movement of the slippers can be approximately calculated to \( \approx 20 \text{ kHz} \) which is distinctly larger than measured oscillations. In addition, the slipper oscillations are pressure dependent, which is not seen in measurements.

**Variable versus fix machine**

The displacement setting valve is locked to ensure constant displacement angle and by this means ensure constant flow. The swash-plate system’s, including the supporting spring, natural frequency is below 100 Hz and should not cause any disturbance in the measurements. If the swash-plate motion were to cause the oscillations, the motion needs to be \( \approx 10 \mu \text{m} \). The swash-plate is supported by a hydrostatic bearing which is pressurised from a small orifice in the piston through the hydrostatic bearing at the slipper and finally a small orifice in the swash-plate. The bearing is only pressurised when a piston is moved over the opening. Also, the force acting on the bearing can not be unambiguously determined. This may be causing the additional oscillations but it is very difficult to prove.

**Machine size**

The volume that separates the dynamic load from the static pressure needs to be sufficiently large to isolate the measurement pipe from the rest of the system. The machines have different displacements, the in-line is bigger, 60 cm\(^3\)/rev, while the bent-axis has 40 cm\(^3\)/rev. The in-line pump is variable and measurement was performed with the same amount of mean flow as for the bent-axis pump. However, the problem with the enhanced amplitude of the higher frequencies remains.
Source flow measurement

Transducer location

It may be difficult to measure the pressure at the creation of the source flow on which the method is based. However, two different locations of the pressure transducers are tested on the bent-axis machine and no significant differences can be seen. The in-line machine’s potential transducer locations are limited and only one spot is tested.

Girder at outlet port

The in-line pump has strengthening supports at the high pressure kidney on the valve plate. The girder can create a turbulent flow and destroy the measured pressure wave. Also, in [45], the pressure between the valve plate and the cylinder barrel is simulated and at the girder, the pressure is raised, which can influence the measurement. The girder was removed but no significant differences could be found.

Cylinder barrel

The cylinder barrel can cause oscillations. In some pumps, the cylinder barrel is supported by a large bearing to reduce the force on the upper part of the barrel. The bearing reduces the noise/vibrations from the pump and may also reduce the flow pulsations.

No clear explanation of the measurement results can be found at this stage. However, the measurement method is suitable for complex outlet channels with fairly low flow pulsations. The main drawback of the method is the need for an additional transducer located on the pump shell.
6 Audible noise prediction

A simulation model used for noise reduction investigation in hydraulic pumps and motors should include characteristics which are considered to create audible noise somewhere in the hydraulic system. In the optimisation chapter, section 4, different objectives were investigated. However, it is difficult to rate the importance of the different characteristics. It may be useful to predict the characteristics’ influence on the audible sound, hence the characteristics are directly valued in noise impact.

This can be achieved by using a transfer function. The transfer function is a black box modelling the pump shell’s excitation capability of audible noise in the frequency domain. The method is based on the idea that some system parameters produce the emitted noise from the machine housing. By modelling a black box between simulated forces and measured audible noise at the machine housing the sound pressure can be predicted. The black box can be used to investigate different design solutions by means of emitted noise.

The method is only valid where the dynamics of the pump can predict the structure borne noise. The sound due to e.g. cavitation, which can appear as a result of a poorly designed valve plate, can not be predicted with the method.

It is not possible to study variations in the machine shell itself because this would also affect the black box. The machine manufacturer does not change the machine housing regularly and it is not unusual to see 40 year old pump housing designs. This approach is relevant for machine manufacturers to investigate new changes in e.g. the valve plate where the force balance is changed.

A more general approach has to be used if the pump housing design is to be developed, e.g. a coupling between finite element model (FEM) of the machine and the dynamic forces in the machine. In [90] the excitation forces are applied to a FEM-model which is used to evaluate the structure noise level.
thus the noise at the pump housing. [123] takes this method one step further. After using the pulsating forces and a FEM-model, the article proposes the boundary element method (BEM) to predict the sound power radiated from the pump housing. [70] shows that modification of the basic structure of the pump housing can reduce the structural noise by 3 dB. All the references above use the internal pulsation forces to predict the structure vibration and thereby the noise radiated from the pump housing.

The idea behind the transfer function methodology is to predict the audible noise excitation from the pump housing. The main noise contribution is considered to be axial piston force and its moments. Flow pulsation also contributes however; the short channel, where the flow pulsations are transformed into pressure pulsations is short and only very small impact on the pump housing vibration is supposed. The flow pulsation has greater impact on the external system compared to the structure vibration at the pump housing. However, the flow pulsation is extremely important to consider because the flow pulsations created at the pump can travel through the pipe line system and excite noise in many different places.

As stated in the conclusion of [66], the emitted noise is dependent on both flow pulsations and oscillated forces in the machine. However, this methodology assumes that the external system is well-insulated and only the pump casing affects the microphone.

The question that the transfer function methodology may answer is: How should the noise reduction feature, such as pressure relief grooves, be designed to minimise structure borne noise emitted from the pump housing?

6.1 Obtain the transfer function

Through the transfer function methodology the sound pressure can be predicted for different valve plate designs and other noise reduction features. In [95], this method is presented and in papers [III] and [IV] is used to investigate optimisation of valve plate designs and non-uniform pump/motor unit respectively. The transfer functions used in the two papers is from [95]. Section 6.3 shows transfer function elaborated from new sound pressure measurements with a slightly different machine housing but still a bent-axis seven piston machine. The measurement technique, which is explained below, is the same as in [95]. The choice of valve plate design is slightly different.

The variables which are supposed to create the audible noise at the machine shell are axial piston force $F_z$ and moment around x and y axis $M_x$ and $M_y$ acting on the valve plate. The forces used are presented in section 2.1.3. There are also other noise sources however; the axial piston force is the largest force and is considered to be the largest creator of sound in normal conditions. Through condition monitoring of e.g. pump shell vibration, faults can be detected at an early stage; the methodology presented here however assumes that there are no faults in the machine or bad work environments. Further discussion about
noise creators in a fluid power machine can be found in section 6.2.

Equation (6.1) shows the equation system to be solved to find the black box model, i.e. $G_{Fz}$, $G_{Mx}$, and $G_{My}$.

\[
\begin{bmatrix}
A_a \\
A_b \\
A_c
\end{bmatrix} = \begin{bmatrix}
F_{za} & M_{za} & M_{ya} \\
F_{zb} & M_{zb} & M_{yb} \\
F_{zc} & M_{zc} & M_{yc}
\end{bmatrix} \begin{bmatrix}
G_{Fz} \\
G_{Mx} \\
G_{My}
\end{bmatrix}
\]

(6.1)

The three transfer functions, $G_{Fz}$, $G_{Mx}$, $G_{My}$, determined in the frequency domain, are needed to describe the audible noise created at the machine shell. To determine the transfer functions, three different combinations of simulated force and moments are needed together with the measured noise quantity. $A$ in equation (6.1) is measured audible noise, $F_z$, $M_x$ and $M_y$ are the corresponding simulated force and moments while $a - c$ indices for the different excitation-response pairs, i.e. the different valve plate designs.

The different combinations of force and moments are realised by using three completely different valve plates. The valve plates are designed to create a large and different force and moment spectrum. It is not obvious how to determine the choice of valve plate design. The determinant can be calculated and the most beneficial valve plate combination can be chosen. The three valve plates used in this study are:

- **Zero-lapped** This valve plate gives rapid transitions between low and high pressure ports and vice versa. The force is almost a square wave in the time domain and gives rise to a broadband frequency spectrum.

- **Exaggerated pre-compression** The pre-compression angle is placed at the connection to the high pressure port at bottom dead centre. The angle is fairly large and the pressure in the cylinder has been built up considerably higher than the pressure in the high pressure port when the cylinder connects to the port. At the top dead centre the valve plate is zero-lapped, i.e. no pre-compression angles. The frequency spectrum is changed considerably compared to the zero-lapped valve plate.

- **Motor** There are pressure relief grooves at all port angles. The spectrum is changed from the other two valve plates.

- **Pump** A valve plate with a well designed pressure relief groove at the entrance to the high pressure port. At the top dead centre the valve plate is zero-lapped, i.e. no pre-compression angles. This valve plate is only used for validation of transfer functions.

The simulated force and moments for the different valve plates are shown in figure 6.1.
When the transfer functions are determined, the predicted sound pressure, $A$, can be calculated with simulated axial piston forces and moments ($F_z$, $M_x$, and $M_y$) for a new valve plate design with equation (6.2).

$$A = G F_z F_z + G M_x M_x + G M_y M_y$$  \hspace{1cm} (6.2)

### 6.2 Different source of noise

There are many different sounds which occur in and around a fluid power machine. The noise is not only annoying and reduces the usefulness of hydraulic power but is also indicative of efficiency losses and fatigue in the products. It is only the pulsating forces or moments which create pressure waves in the air. Therefore, the interesting part is the pulsating amplitude of the force or moment.

Most of the noise that occurs is proportional to the forces created in the machine, i.e. the pressure which the machine works at. A short explanation of
the different noises is given below. The discussion is formulated according to what sound pressure occurred at the machine housing which means the machine housing is the only exposed structure for the sound pressure transducer. The connecting structures such as pipe line systems and drive motor are screened off. The noise can be divided into external and internal parts.

External noise

**Axial forces** The predominant source of the structure borne noise from the machine housing is the forces created due to the pressurisation inside the machine. The noise produced is proportional to the force, i.e. pressure. The axial piston force and the moments on the valve plate is calculated according to equation (2.12)-(2.16).

**Radial forces** Radial forces appear mainly as synchronisation forces and piston radial forces. The total moment at the synchronisation feature is a fraction of the total transferred moment. The piston radial force is negligible compared to the piston axial force. The noise resulting from the axial forces is proportional to the pressure level.

**Torque** The moment around the drive shaft is calculated as equation (6.3) for a bent-axis machine.

\[
M_{\text{axis}} = \pi R_p^2 \sum_{j=1}^{z} (-R_d p_{c,j} \sin \alpha \cos \phi_{c,j} p_{c,j})
\]  

(6.3)

The noise from torque is properly predominant at the mounting and the construction connected to the pump housing.

**Flow pulsations** Flow pulsations are created at the valve plate and are called source flow. The flow pulsation interacts with the inlet and outlet channel of the pump and produces pressure ripple. The pulsation is spread through the pipe line system to a completely different place than the pump. This noise must obviously be kept in mind while designing quiet machines.

However, the channels in the pump are fairly short and the impact on the noise at the machine housing is presumably of minor interest compared to the axial piston forces, provided that the external pipe system is well isolated. The flow pulsations are proportional to the force as long as the piston number and the symmetry are kept unchanged.
Internal noise

**Synchronisation** Mainly involves bent axis machines with synchronisation between rotation axis and cylinder barrel. The connection may be of different kinds but is assumed to be proportional to the force.

**Bearings** The noise level from bearings involves for example what type of bearing is used, the precision of the geometric forms and surface as well as type of lubricant. For a given machine with the same condition such as oil the bearing noise is assumed to be proportional to the force and therefore considered through the axial piston force.

**Flow** Flow noise is caused by the oil in the machine moving precipitously or fast. One flow noise appears as jet flows, which are created at the valve plate opening due to the high speed of the oil. Cavitation appears when the oil pressure is reduced to below the saturation pressure and vacuum filled bubbles appear in the oil. Subsequently, when the pressure increases, the vacuum bubbles implode and cause a high frequency noise and major wear problems.

Churning noise is probably most apparent at high speed and is low compared to other noise. In [118] a method to reduce churning losses is presented and the insert feature can give 3-5 kW more power. Flow noise stated above is all not proportional to pressure, which implies that the sound pressure measurement method does not work for severe flow noises in a machine.

6.3 Measurement approach

Sound pressure measurements are performed in a semi-anechoic room. The room is dimensioned according to ISO standard where the prime-mover is located outside the measurement room. However the load, i.e. the anechoic termination system, is placed inside. Figure 6.2 shows the hydraulic system for the measurements.

The test pump is driven by a hydraulic motor with a fly-wheel connected on their common axis. The fly-wheel is used to keep the rotational speed constant during measurements. The Fourier analysis becomes meaningless if the rotational speed changes during data acquisition. The inertia of the fly-wheel is 7 kgm² which is well above the expected needs. The fly-wheel and the rest of the drive system are placed outside the semi-anechoic room. The parts which are not possible to mount outside are well protected from the microphone with insulation material. Only the pump housing is exposed to the microphone. Figure 6.3 shows the semi-anechoic room with the test object and microphone.

All measurements are made with anechoic conditions at the pump outlet channel to eliminate wave reflections in the pipeline. The sound pressure me-
Figure 6.2 Measurement set-up to obtain the transfer functions. The lines shows which features are inside the semi-anechoic room. The absorbent material is placed so as to protect additional parts of the measurement system to be exposed to the microphone. This is shown with the bold dashed lines.

Figure 6.3 The semi-anechoic measurement room with only the test object exposed for the microphone.

ter is placed 1 metre right in front of the pump housing. The transfer functions are only valid for the specific position where the measurement has been performed. The sound pressure signal is sampled at 200 kHz using a second order Bessel-type anti-aliasing filter. The cut-off frequency chosen is 10 kHz. The resulting frequency resolution is approximately 1.5 Hz. The sound pressure measurements and hydraulic pressure is sampled simultaneously.

The simulations and measurements have to be synchronised and this is done through flow pulsation measurements. All the measurements are performed
under anechoic conditions and hence the flow pulsations can be measured directly through one pressure transducer. The choice of measurement technique is justified by the selected pump model. The source flow is also an output signal from the simulation model.

To ensure accurate determination of the transfer functions, the Fourier series coefficients are validated and only the good frequencies used. The standard deviation, equation (6.5), of the harmonics mean value, equation (6.4), is calculated. The limit is set at 20% of the amplitude of the mean value. The coefficients are complex values and hence there are some different approaches to calculate the standard deviation.

\[
M = \frac{\sum_{i=1}^{n} p_i}{n} \tag{6.4}
\]

\[
s = \sqrt{\frac{\sum_{i=1}^{n} |p_i - M|^2}{n-1}} \tag{6.5}
\]

The pump’s rotational speed when determining the transfer functions is between 900-2,000 rpm with steps of 50 rpm. The resolution of the determined transfer functions is 20 Hz. The frequencies at all rotational speeds that fall within the range of 20 Hz will be considered when calculating the transfer functions. If the system becomes over-determined, the system is calculated in the least square sense. Conversely, if the number of equations falls below three the frequency interval will be empty. Linear interpolation is used to find the intermediate values when predicting the sound pressure level.

### 6.3.1 Results

The obtained transfer functions are shown in figure 6.4. The amplitude of the transfer functions clearly shows the sound pressure behaviour at increased frequency. The transmission from forces to sound pressure is higher at 1,500 Hz than 200 Hz. Also, the transfer function representing axial piston force has greater impact compared to transfer functions representing moments. When predicting the sound pressure by using the transfer function the phase is equally important. Figure 6.4(b) shows the phase of the transfer functions. The phase shift is reduced in the figure. There is a large phase shift due to the microphone being placed one metre from the source.

Verification of the transfer functions is shown in figure 6.5 in both the time and the frequency domains. The figure shows the predicted sound pressure and the measured sound pressure. The measurements are made with a pump valve plate with a pressure relief groove at the cylinder connection to the high pressure port. The transfer function values can be compared to the transfer
functions in [95]. The pump shell is similar and it is no surprise that the transfer functions obtained are similar. Also, the measurements and hence the transfer functions will always be dependent on the environment. It would have been beneficial to increase the upper frequency limit due to the increased sound pressure. The main limitation for higher frequencies is that the standard deviation becomes large and hence the results become unpredictable.
6.5(a) 1,700 rpm and 200 bar

6.5(b) Left: 1,800 rpm and 200 bar and right: 2,000 rpm and 200 bar

Figure 6.5 Measured sound pressure, dots and dashed line, and calculated sound pressure, bars and solid line, at different rotation speeds.
As stated in chapter 2 there are many different ways to reduce noise in hydraulic pumps/motors. In this thesis two different noise reduction interventions are tested. Section 7.1 investigates the possibility to reduce noise by placing the pistons and/or the cylinder kidney angle in a non-uniform way. The amplitude of the harmonic frequencies in a uniform pump/motor may then be reduced by moving the energy content to other frequencies. The investigation is only made for a fixed bent-axis pump.

In section 7.2 a so called cross-angle is tested for a pump/motor working with both positive and negative displacement angles. The cross-angle changes the top and bottom dead centre and thus the size of the pre- and decompression angles. The cross-angle was patented 1963 [17] and some early references are [7, 43]. In later years, the feature has been investigated and experimental verified in [54–56]. The cross-angle for pump/motor units are investigated in paper [I] and are further studied in the thesis [X]. The investigated unit is a variable bent-axis machine with relatively large maximum displacement.

### 7.1 Noise reduction by means of non-uniformity

In an ordinary uniform machine, the harmonic frequencies depends on the rotational speed and multiple of the number of pistons, equation (7.1). In a seven-piston machine the energy content is located at every seventh harmonic
frequency, i.e. \( i = 7, 14, 21, \ldots \)

\[
\omega = 2\pi ni \quad \left\{ \begin{array}{ll}
\text{uniform machine} & i = 1z, 2z, 3z, \ldots \\
\text{non-uniform machine} & i = 1, 2, 3, \ldots 
\end{array} \right.
\] (7.1)

A non-uniform machine means that there is only one period per revolution instead of number of pistons. Thus, the amount of harmonic frequencies which may contain energy increases by factor \( z \). The pulsations’ periodicity is thus changed, which can have a major impact on the noise level and how the noise is experienced. It is not obvious how a human ear perceives the noise and how the pump housing interprets the additional frequencies.

There are a number of benefits which non-uniformity can bring:

- **Lower the fundamental frequency**
  For a seven-piston pump driven at 2,000 rpm the fundamental frequency becomes \( \approx 233 \) Hz for a uniform machine while in a non-uniform machine the fundamental frequency becomes \( \approx 33 \) Hz. The human ear is less sensitive to lower frequencies and this may be an advantaged in a non-uniform machine.

- **Decrease the maximum amplitude**
  The energy content can be spread to nearby frequencies and hereby the maximum amplitude of single harmonic can be decreased. However, the total energy is most probably increased by the non-uniformity.

- **The human ear experience**
  The non-uniformity will change the experience of the noise, however this is probably a very subjective parameter to consider. One way to consider the interpretation of a human ear is to use the A-weighted sound pressure level value of the noise at the pump housing.

There are many possible methods to make the machine non-uniform. The easiest from a production point of view is to move the cylinder kidney, i.e. the opening at cylinder barrel which connects the cylinder and valve plate ports. The main change in the flow ripple is the compressible parts due to the change in pre- and decompression angle. The other three modifications considered concern the pistons: Individual piston radius, piston pitch angle, and radial displacement of the pistons.

An angular displacement of one cylinder kidney is used to validate the simulation model. The other methods are modelled in the same way and hence no validation is needed.

### 7.1.1 Optimisation of the non-uniformity

Simulation based optimisation is used to investigate the usefulness of the non-uniform machine. Four different optimisation objective functions are considered.
Noise reduction in axial piston machines

1. The amplitude of the flow pulsation in the time domain. This is probably the most common characteristic to consider when reducing noise in fluid power machines.

\[ f_1(\Psi) = \max(q(t)) - \min(q(t)) \]  
\[ \text{(7.2)} \]

2. Minimise the maximum amplitude of the flow pulsation spectra which is 7th or 14th harmonic in a uniform machine. The reduced energy at this harmonic is moved to nearby harmonics and also to the first harmonic frequency.

\[ f_2(\Psi) = \max(Q_k(j\omega)) \quad \text{where} \quad k = 1 \]  
\[ \text{(7.3)} \]

3. The total energy content is most probably increased by the non-uniformity and this objective function investigates if it is possible to reduce the total energy content of the flow pulsation in any of the considered modifications.

\[ f_3(\Psi) = \sum_{k=1}^{n} Q_k(j\omega) \]  
\[ \text{(7.4)} \]

4. The objective function minimises the audible noise at the pump housing. The audible noise is predicted through the transfer function methodology [95]. The predicted sound pressure is A-weighted to simulate interpretation of the audible noise by a human ear.

\[ f_4 = 10 \cdot \log \left( \sum_{n=1}^{N} 10^{\frac{L_{p n} + \Delta_{an}}{10}} \right) \]  
\[ \text{(7.5)} \]

Four different pressures are used in the optimisation \( p = 5, 15, 25, 35 \) MPa and both the high and low pressure port flow pulsation when applicable, i.e. \( f_1, f_2, \) and \( f_3 \).

The rotational speed is kept constant during the optimisations. The results will have the same relative values for objective functions \( f_1 - f_3 \). The fourth objective function which involves the sound pressure level, will be very dependent on the amplitude of the transfer function harmonic and hence the investigation was limited to a fixed rotation speed. The optimisation will search for the design parameters which minimise and maximise the harmonics where the measured transfer functions are smallest and largest at the different rotation speeds. This investigation is probably more efficient and analysable in a simulation environment like an FEM model.

The design variables are parameters which are connected to the modification, i.e. when the cylinder kidney pitch angle is optimised only the pitch angles are used as design variables and so forth. The valve plate is fixed during all optimisations to an ordinary manufactured valve plate used in machines which work only as pumps. The mean flow is restricted to the same value as the uniform pump and the cylinder walls are not allowed to be too thin.
7.1.2 Optimisation results of non-uniform pump

The optimised results for the different approaches are shown in table 7.1 together with the objective function values for the uniform original pump. Cylinder kidney pitch angle is not quit comparable with the other results due to the change of pre- and decompression angles.

It is hardly not possible to reduce the amplitude of flow pulsations in time domain $f_1$ or the sum of the produced energy, $f_3$ with any of the suggested methods. The maximum harmonic of the flow pulsation can be reduced by the piston’s pitch angle $f_2$ but all the other investigated objective function show a significant increase. Also, the audible noise at the pump housing $f_4$ is reduced by the pistons’ pitch angle but there is a major drawback in other objective functions. It must be remembered when studying the sound pressure level in the unit dB that an increase of 3 dB represents a power which is doubled. However, a doubling of the perceived sound level requires an increase of 8-10 dB.

<table>
<thead>
<tr>
<th>Design variable</th>
<th>Obj. Func.</th>
<th>$f_1$ (l/min)</th>
<th>$f_2$ (l/min)</th>
<th>$f_3$ (l/min)</th>
<th>$f_4$ (dB(A))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform machine</td>
<td></td>
<td>14.00</td>
<td>2.33</td>
<td>7.39</td>
<td>64.19</td>
</tr>
<tr>
<td>Pistons’ pitch angle</td>
<td>$f_1$</td>
<td>13.97</td>
<td>2.33</td>
<td>7.76</td>
<td>64.19</td>
</tr>
<tr>
<td></td>
<td>$f_2$</td>
<td>18.19</td>
<td>2.21</td>
<td>16.14</td>
<td>64.48</td>
</tr>
<tr>
<td></td>
<td>$f_3$</td>
<td>14.02</td>
<td>2.33</td>
<td>7.49</td>
<td>64.19</td>
</tr>
<tr>
<td></td>
<td>$f_4$</td>
<td>16.84</td>
<td>2.22</td>
<td>14.83</td>
<td>64.03</td>
</tr>
<tr>
<td>Cylinder port pitch angle</td>
<td>$f_1$</td>
<td>10.94</td>
<td>2.03</td>
<td>6.09</td>
<td>64.53</td>
</tr>
<tr>
<td></td>
<td>$f_2$</td>
<td>10.94</td>
<td>2.03</td>
<td>6.16</td>
<td>64.54</td>
</tr>
<tr>
<td></td>
<td>$f_3$</td>
<td>13.99</td>
<td>2.03</td>
<td>6.06</td>
<td>64.52</td>
</tr>
<tr>
<td></td>
<td>$f_4$</td>
<td>12.31</td>
<td>2.15</td>
<td>6.65</td>
<td>63.62</td>
</tr>
<tr>
<td>Radial displacement of the pistons</td>
<td>$f_1$</td>
<td>14.00</td>
<td>2.33</td>
<td>7.51</td>
<td>64.19</td>
</tr>
<tr>
<td></td>
<td>$f_2$</td>
<td>14.97</td>
<td>2.33</td>
<td>8.33</td>
<td>64.20</td>
</tr>
<tr>
<td></td>
<td>$f_3$</td>
<td>14.00</td>
<td>2.33</td>
<td>7.48</td>
<td>64.19</td>
</tr>
<tr>
<td></td>
<td>$f_4$</td>
<td>14.34</td>
<td>2.33</td>
<td>7.85</td>
<td>64.19</td>
</tr>
<tr>
<td>Piston area</td>
<td>$f_1$</td>
<td>13.99</td>
<td>2.33</td>
<td>7.49</td>
<td>64.19</td>
</tr>
<tr>
<td></td>
<td>$f_2$</td>
<td>17.76</td>
<td>2.33</td>
<td>12.06</td>
<td>64.28</td>
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<tr>
<td></td>
<td>$f_3$</td>
<td>14.00</td>
<td>2.33</td>
<td>7.48</td>
<td>64.19</td>
</tr>
<tr>
<td></td>
<td>$f_4$</td>
<td>18.05</td>
<td>2.42</td>
<td>12.40</td>
<td>64.12</td>
</tr>
</tbody>
</table>
Figure 7.1 shows the percentage displacement of the pistons' pitch angle when the four objective functions are used. Only objective functions $f_2$ and $f_4$ show a major change from a uniform pump. Regarding the objective function $f_4$, it seems as if the optimisation tries to make the axial piston force as smooth as possible.

![Figure 7.1](image_url)

**Figure 7.1** The change in the angular position of the pistons in percent from an uniform placement, i.e. $2\pi/7$. Objective functions $f_1$, $f_2$, $f_3$, and $f_4$ are shown from left to right and from light grey to dark grey.

The reduction of the maximum amplitude of the flow pulsation harmonic spectra is mainly done by the low pressure flow pulsation, shown in figure 7.2. The figure also shows the total axial piston force frequency spectra and the force point of action.

In addition to the optimisation presented above, combinations of the modifications and objective functions can be considered. If all four objective functions are used in a multi-objective optimisation operation the result becomes a uniform machine design. The noise level is reduced by less than 0.2 dB(A) if the pistons' pitch angle and the radius and the radial displacement of the pistons are simultaneously used as design variables. However, this optimisation is very complex because of the non-linear behaviour of the objective function and all the dependent design variables which will be included. The risk of finding a local optimum is great; hence, the optimisation may not be unequivocal.

The specific design which was optimised by the sound pressure level is dependent on the rotational speed and the pump housing design and will most probably change if these conditions are changed. In addition to the rotational speed shown above some other rotational speeds were investigated and show similar results both in the design variable and objective function values. One aspect which is not considered by using an A-weighted filter is the reduction by the largest amplitude of the individual harmonics of the sound pressure,
Figure 7.2 The results when the piston pitch angles are optimised with objective function $f_4$ equation (7.5), solid line and filled bars. Dashed line and empty bars show a uniform machine.

see [15] among many others. The noise can be more pleasant when the energy content is spread to more frequencies.

There are other approaches to make the machine non-uniform which could be of interest to investigate. Some approaches are specific to some pump configurations and not as general as the approaches presented here. Also, the results show the complexity of making a non-uniform machine good in different characteristics simultaneously.
7.2 Cross-angle for pumps and motors

The cross-angle is a fixed displacement angle, $\gamma$, perpendicular to the original displacement angle, $\alpha$. This results in an adjustment of the dead centre angle $\eta$, as a combination of the normal displacement angle and the cross-angle, according to equation (7.6).

$$\eta = \arctan \left( \frac{\tan(\alpha)}{\tan(\gamma)} \right)$$  \hspace{1cm} (7.6)

Figure 7.3 shows different swash-plate inclinations at positive and negative displacement angles. A pump’s resulting inclination, when cross-angle is implemented, is shown in figure 7.3(c). The dead centre angle $\eta$ decreases when
On Fluid Power Pump and Motor Design

the displacement angle $\alpha$ decreases. A machine which can have negative displacement angles with the same high pressure kidney and rotational direction will operate as a motor; the obtained inclination, including a cross-angle, is shown in figure 7.3(d). The dead centre angle increases when $\varepsilon$ goes from -1 to 0. The movement of the dead centre location results in additional pre- and decompression in both pump and motor modes, which is needed for optimal pressure equalisation at small displacement angles.

The same inclination direction can be used for both clockwise and anticlockwise rotational direction and both positive and negative displacement settings. All these four quadrants can be used with preserved beneficial influences from the cross-angle. However, the pressure ports cannot be swapped if the same inclination direction of the cross-angle is to be used. For such machines, the cross-angle inclination therefore has to be variable. The inclination direction should slope towards the low pressure side.

7.2.1 Optimisation of the cross-angle and valve plate design

The optimal cross-angle does not have the same value for all the quadrants and also the high and low pressure port flow ripple; a trade-off therefore needs to be found. To find the trade-off and hence the optimum cross-angle to minimise the flow ripple in both the high and low pressure ports a Pareto front is created. The Pareto front for both pump and motor mode are shown in figure 7.4 at 25 MPa and 2,000 rpm.

The different modes are realised by changed displacement direction, i.e. $\varepsilon$ is -1, -0.5, and -0.05 for motor optimisations and $\varepsilon$ is 1, 0.5, and 0.05 for pump optimisations. Every size of cross-angle and also the quadrants need special valve plate timing in the optimal design and hence a compromise has to be made for a final machine design.

The size of the cross-angle diverges between pump and motor mode. To minimise high pressure flow ripple in the pump case the cross-angle should be rather small while in the motor case the size should be rather large and vice versa for the low pressure flow ripple. However, the total flow ripple, i.e. flow ripple in both high and low pressure ports, minimisation is obtained at $\approx 4^\circ$ in both the pump and motor cases for the particular machine that is simulated and also the operating condition. The pre- and decompression angles vary with different cross-angles and driving modes. However, the best combination of the valve plate timing coincides with the cross-angle that minimises the total flow ripple, $\gamma \approx 4^\circ$. This is further illustrated in figure 7.5 for another operating condition.

Approximately, the best compromise valve plate timing for the combination of the pump and motor mode at 35 MPa coincides with the cross-angle where the total flow ripple minimises, which is at about $\gamma \approx 5^\circ$. In figure 7.4, the cross-angle was optimised for pressure at 25 MPa and the best compromise cross-angle size and valve plate timing was $\gamma \approx 4^\circ$. 

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Figure 7.4 Pareto optimal fronts for pump case (circles) and motor case (crosses). Pareto optimal front illustrating the trade-off between the flow ripple in the high and low pressure kidneys. The x-axis shows the average peak-to-peak value of the high pressure flow ripple at three different displacement angles, whereas the y-axis shows the mean peak-to-peak value of the low pressure flow ripple. The arrows indicate the values of the optimised cross-angle at different locations on the Pareto front.

Figure 7.5 Left: Kidney angles for high pressure opening angle and low pressure closing angle when the cross-angle varies. Right: Kidney angles for high pressure closing angle and low pressure opening angle when the cross-angle varies. Solid lines are for motor mode and dashed lines pump mode.
The flow ripple as a function of cross-angle is shown in figure 7.6. The arrows show how the optimal cross-angle changes when rotational speed is increased and decreased and also variation in system pressure. The rotational speed variations do not affect the size of the cross-angle. This is logical because the pre- and decompression angles are fairly insensitive to rotational speed variations. The pressure variations affect the cross-angle. Greater pressure needs larger pre- and decompression angles and this has an impact on the cross-angle; the optimal cross-angle for both the motor and pump mode is bigger at greater pressures.

### 7.2.2 Rating the use of cross-angle

There is no obvious weighting function for the different harmonics created in machines. Any of the commonly used weighing methods will have its pros and cons. In this thesis the Root Mean Square function (RMS) is used to obtain comparable figures of merit for the flow ripple created in the machine, according to equation (7.7). The cross-angle is $\gamma = \gamma_0^\circ$ and the pre- and decompression angles are a compromise according figure 7.5.

$$Q_{\text{RMS}} = \sqrt{\sum_i \sum_j \sum_{\varepsilon} (Q_{ij\varepsilon})^2} \quad \text{where} \quad i = \text{harmonic frequencies} \quad j = \text{high, low} \quad \varepsilon = 0.05, 0.5, 1$$

In figure 7.7, the RMS-value is shown when only the harmonics are summarised; displacement settings and low and high pressure flow ripple are sepa-
rated. The flow ripple is decreased at all the displacement angles in the pump mode and in both the high and low pressure port, figures 7.7(a) and 7.7(b).

![Graphs for pump mode - high and low pressure](image)

**Figure 7.7** The RMS value of the flow ripple at different displacement settings and in both ports. The squares show when no cross-angle is implemented and a zero-lapped valve plate is used. Points show the results when a cross-angle is implemented.

In the motor mode, shown in figures 7.7(c) and 7.7(d), the flow ripple is decreased in the low pressure port at all the displacement settings but in the high pressure port the flow ripple is increased for large displacement settings.

A complete use of equation (7.7) gives a 40% reduction in the flow ripple when a cross-angle is implemented compared to a zero-lapped valve plate when no cross-angle is used.

In the analysis above a cross-angle which minimises the total flow ripple is used, but it may not be the best choice for all applications. If a bigger cross-angle is chosen for the motor-case, the high pressure flow ripple is decreased at the expense of the low pressure flow ripple. This can be justified if the high pressure flow ripple tends to spread in the system more than the low pressure flow ripple.
7.2.3 Complementary features together with the cross-angle

In the analysis above a fixed cross-angle is considered. If it was possible to make the cross-angle variable, it is difficult to find a good valve plate timing to suit different cross-angles at different pressure levels to minimise the flow ripple. However, it is a demand to have variable cross-angle with different directions when the pressure side is changed on the machine. As stated above, the cross-angle inclination can be combined with other features to reduce the pressure dependency. The two tested features are pre-compression filter volume (PCFV) and pressure relief groove implemented before the high pressure kidney. The features have slightly different optimal dimensions for the pump and motor mode, caused by the motion of the piston and the cylinder volume in the pre-compression zone. The dimensions of the features are not optimised with any numerical algorithm, but the sizes are nevertheless well suited to the application. The cross-angle is set to $4^\circ$, which is optimal for ordinary pre- and decompression angles at 250 bar. All the features are modelled with the same boundary conditions: maximum overshoot 5 MPa, no cavitation allowed, which means the low pressure is not allowed to fall below 1 bar any time during the simulation, no leakage between the kidneys.

Pump mode

Figure 7.8(a) shows the sensitivity to pressure and displacement angle variations when a cross-angle is used with only ordinary pre- and decompression angles in pump-mode. The other features in combination with a cross-angle are shown in figure 7.8(b) for pressure relief groove implementation while the PCFV is shown in figure 7.8(c). The results show a decrease in sensitivity to pressure variation when a pressure relief groove is used and an even better insensitivity to pressure variations when a PCFV is used. The results from these simulations can be compared to a pump with no cross-angle, shown in figure 7.9. The pump with pre- and decompression angles and no cross-angle can not be used in the motor mode because of cavitation and pressure overshoots. All other designs can be used in both pump and motor mode.

Motor mode

Figure 7.10(a) shows the optimised result for the motor application when cross-angle and ordinary pre-compression angles are used. Cross-angle in combination with pressure relief groove is shown in figure 7.10(b) and cross-angle combined with PCFV is shown in figure 7.10(c).

There are improvements when the additional features are used, however the improvements were more significant for the pump mode. Both cylinder and pre-compression volume must therefore be loaded by the high pressure kidney for optimal use of the features; therefore, the pressure relief groove and the PCFV therefore have to be placed at the entrance to the high pressure kidney,
not at the pre-compression zone in the motor case. This means that the feature is located at the top dead centre. This makes the features less efficient for noise reduction.
Figure 7.8 The flow ripple’s sensitivity to pressure and displacement setting variations in pump mode when cross-angle is used.
Figure 7.9 The flow ripple’s sensitivity to pressure and displacement setting variations in pump mode, no cross-angle implemented.
Figure 7.10 The flow ripple’s sensitivity to pressure and displacement setting variations in motor mode when cross-angle is used.
Figure 7.11 The flow ripple’s sensitivity to pressure and displacement setting variations in motor mode.

7.11(a) Zero-lapped valve plate

7.11(b) Pre- and decompression angles
Conclusions

The complexity of reducing the noise created in a fluid power pump/motor unit has been shown in the project. Even so, the thesis extends to just examine the pump/motor unit as an autonomous part and tries to limit the interaction with the environment. However, the thesis treats the results in terms of systems that a pump/motor might be part of. To be able to investigate the noise contribution from pump/motor units in a useful effective way, good all-round tools are needed. The thesis discusses how the objective function in the optimisation algorithm should be formulated. Many of the results are from simulation based optimisations where simulation software in combination with an optimisation algorithm is used to investigate new noise reduction features. The simulation results are of less importance if the used model is not extensively validated and hence measurement has been an important part of the study. The model should include the important features for the purpose of the model, neither more nor less. CFD calculation has improved a lot the last year and the simulation shows good visualisations of flows and pressures. However the calculation still takes a lot of time. CFD can increase understanding of the flow and a substitute for real experiments, which is expensive and time-consuming.

Unsteady flow through a restrictor sometimes creates additional oscillations, which the ordinary static orifice equation fails to describe. If the model also take into consideration the inertia effect of the jet formed in the orifice a much better agreement with experiments is obtained. The model has one extra parameter which is obtained from CFD calculations or experiments. The extended model thus explains the discrepancy between simulations with established models and measurements, in case of rapid orifice area changes. When the area gradient is reduced by means of pressure relief groves, this inertia effect is reduced and can mostly be neglected.

An important issue when considering noise contributions in fluid power pump/motor is the amount of free air in the oil. The effective bulk modulus of the oil decreases drastically when the free air content increases. The
pressure difference between the low and high pressure port is one parameter to consider. The probable cause of this is the jet-stream that is created when the pre-compression is insufficient. The same theory may also be a plausible reason why the pre-compression valve plate creates somewhat less free air than for a zero-lapped valve plate. Hence, different valve plate designs affect the amount of free air. If the amount of free air is measured both at the inlet and outlet of the pump, the amount of free air created inside the pump can be determined.

When minimising the noise contributions to the environment many different objectives have to be considered. There are two main types of noise from fluid power pumps/motors: Structure borne noise and fluid borne noise. Both have to be considered when an all-round quiet machine is to be developed. However, if the pump/motor is to be placed in a specific system the objectives can be weighted differently to match the system requirements. It is also not evident how the different objective should be ranked because the observed noise level is strongly dependent on the system in which the pump/motor is to be used. All the simulation based optimisations performed in this thesis are independent of the external system.

The cylinder pressure derivative is the best overall objective, although both cylinder ports have to be considered. In the pump case, these two objectives are relatively independent of each other. In the motor case, the objectives are more dependent. The criteria for an all-round objective is that the objective should give good reductions in all other considered objectives. The total axial force in the frequency domain is another all round objective which can be considered alone. This objective probably attributes more to the structure borne noise and hence the objective function can be formulated as a multi-objective optimisation together with flow pulsation.

Another suitable objective is to use the air borne noise directly. The technique implies that three transfer functions between force and audible noise are generated from measurements and simulations. Hence, it is possible to predict the audible noise at the pump housing for new valve plate designs by simulating the forces created at the valve plate. The transfer functions are defined by the pump housing structure and also the surroundings in which the sound pressure measurement were performed. It is shown that the flow pulsation has no major impact on noise at the pump housing. The method is rather time consuming and puts high demands on the measurement equipment. How much the audible noise can be reduced is highly dependent on the design of the pump housing.

The transfer function method is used to investigate the possibility to reduce audible noise with pressure relief groove and pre-compression filter volume. By adding a pre-compression filter volume, the sound pressure level can be reduced by at least 2 dB(A) compared to a pressure relief groove. In contrast, if the audible noise is minimised with the filter volume the flow pulsation is increased compared to pressure relief groove. With a carefully selected weighting factor between flow pulsation and audible noise, the pre-compression filter volume has greater potential than a pressure relief groove. In general, the volume of the
Conclusions

PCFV should be bigger when designing a pump/motor with minimised audible noise than for flow pulsations.

The source admittance method measurement technique has been investigated. The method needs an additional pressure transducer mounted at the pump housing compared to other similar measurement methods. The benefit of this method compared to e.g. the two microphone method is that a mathematical model of the outlet channel is not needed. The model can be very complex for some pumps. The results show an oscillating behaviour at about 2000 Hz independent of rotational speed and pressure. These oscillations are not present in the simulation model. It is shown that neither the complexity of the shape of the outlet channel nor the location of the pressure transducer causes the behaviour. The oscillation can not be explained by the internal mass at the valve plate opening. No final conclusions can be drawn so far for the oscillating behaviour in the measurement results when using the source admittance method.

Within the project framework, two different noise reduction arrangements have been investigated. The cross-angle can be used for all variable displacement axial piston pumps/motors with fixed pressure ports. The same inclination direction can be used for both negative and positive displacement angles. The size of the inclination and also the pre-compression angles are not the same for all the operating conditions but a good compromise can be found.

In general a pre-compression angle and similar features increases the amplitude of the first harmonic in motor mode. This has nothing to do with cross-angle itself but the reduction and increase in the compressible respective kinematic flow pulsation. Furthermore, the cylinder pressure rate in the pump decreases when pre- and decompression angles are used and this results in less structural vibration. The overall flow ripple is reduced considerably when implementing a cross-angle.

The feature is most probably easier to implement in in-line pump/motor compared to bent-axis pump/motor. The cross-angle can also be used in combination with pre-compression filter volume and pressure relief grooves to improve sensitivity to pressure variations.

The second noise reduction approach is to make the pump/motor non-uniform. The pistons and/or the cylinder kidneys are placed non-uniformly on the cylinder barrel. In this way the energy can be spread to more frequencies and also the behaviour of the noise change. Non-uniformity did not show great improvements but can be good in special cases, especially when lower flow pulsations are important. Other pump housings may have greater impact on the noise level and move the energy content to nearby frequencies. The behaviour and perception of the noise level may be an advantage. Reduction of the biggest amplitude in flow pulsation can be useful; however, it is difficult to validate the benefits from the modification.
Fluid power pump/motor developers face a very interesting future. Different new operational categories have found the usefulness of fluid power, which requires new products.

One part of the noise reduction area in hydraulic pumps/motors is the air content and even more importantly cavitation. Cavitation has been avoided in this study by increased inlet pressure. However, cavitation and air release are of great importance when designing a pump/motor and are also difficult to avoid in real systems.

In the area of this project there are some interesting future ideas. An attractive extension of the transfer method is to consider the pulsating torsion torque, which occasionally contributes to system noise. This is supposed to mainly influence the noise contribution from structures connected to the pump/motor. In this respect it is comparable to flow pulsations which also contribute to noise via connected impedances. The transfer method could be tested for other types of pumps/motors, e.g. line pump, which have another type of mechanical structure and might produce a different kind of audible noise at the pump housing. However, due to the complex and high demands on the measurements, it might be interesting to connect the force and moments simulated in the 1D model into an FEM-model of the pump housing. In this way the transfer functions are more controlled. Also, the pump housing can be easily changed and optimised. The pump housing design can probably be designed in such a way that some frequencies of the vibration can be cancelled out. In combination with the non-uniform pump designs, this might be very useful.

The non-uniform machines can be further investigated with other possible solutions. For example, the pistons trajectory path which possibly can reduce the kinematic flow pulsation.
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Review of papers

Paper I

Fluid Pulsation Reduction for Variable Displacement Motors by Using Cross-angle

The focus in most research papers is on hydraulic pumps which are usually taken to be the biggest source of fluid borne noise, but hydraulic motors are also a fluid borne noise creator. One feature with no movable parts which changes the design of the pump according to working condition is the cross-angle. Earlier investigations of cross-angle have been made for variable displacement pumps but the same feature can improve fluid pulsations in motors as well.

This paper considers using the cross-angle in variable displacement motors. The cross-angle is a fixed displacement angle around the axis perpendicular to the normal displacement direction. Because of the cross-angle, the pistons dead centres vary as a function of the displacement angle; in this way the valve plate timing is varied and different pre-compression and decompression angles are achieved. A non-gradient optimization technique, the Complex method, is used together with extensive simulation models in order to find the optimal cross-angle for a variable displacement motor. The paper shows that the cross-angle can be used to reduce noise in variable displacement motors. One thing that makes the motor application more difficult is the dependence between outlet and inlet flow ripple, which is not found in pump applications.

Paper II

On Optimal Design of Hydrostatic Machines

In the paper, the simulation model of a fluid power pump is used together with a non-gradient optimization method in order to find the best possible design. A vital part when using optimization to support design is always to formulate
the objective function. Different sources of noise need to be considered when
the objective function is formulated. For example, a design that minimizes flow
pulsations in the suction port will surely perform badly in some other objective.
Therefore, noise minimization could be looked upon as a typical multi-objective
optimization problem.

Traditionally, simulation of machine performance is conducted in the time
domain, but the human ear hears noise in the frequency domain and perceives
high and low frequencies differently. This makes it natural to formulate the
objective function in the frequency domain, which raises the question of how
the different harmonics should be ranked.

In the paper, a number of different approaches to formulate the objective
function are presented and evaluated. The goal of the study is to find a good
all-round objective function which reduces both flow pulsation and structure
vibrations. The objectives considered are flow pulsation in both discharge
and suction ports, as well as pulsation in piston forces and bending moments.
Furthermore, the objectives are studied in both the time and the frequency
domains. The paper found that the best all-round objective is pressure rate in
the time domain.

Paper III

Optimisation of Structure Borne Noise and Fluid Borne Noise
from Fluid Power Pumps and Motors

In this paper, a transfer function methodology is employed for mapping simu-
lated internal pump dynamics, such as piston forces and bending moments, onto
structure borne noise. Using these transfer functions, it is possible to predict
how, for instance, changed valve plate timing affects simulated piston forces
and bending moments and in turn how this will affect audible noise. Hence, it
is possible to design an objective function that directly reflects audible noise.

The paper shows how both structure borne noise and fluid borne noise can
be considered using multi-objective optimisation. Also, pressure relief groove
and pre-compression filter volume are optimised with the transfer function
method. It is shown how certain parameter values are chosen according to
whether audible noise or flow pulsation is to be minimised. With a correctly
chosen size of the volume of the pre-compression filter volume, both the audible
noise and flow pulsation are reduced more compared to pressure relief groove
implementation. The paper considers the difference between pump and motor
cases.
Paper IV

Noise Reduction by Means of Non-uniform Placement of Pistons in a Fluid Power Machine

This paper discusses the possibilities to introduce non-uniform placement of the pistons. The pulsations’ periodicity is thus changed, which can have a major impact on the noise level and how the noise is experienced. A number of approaches are presented, evaluated and ranked and the usefulness of the modifications is assessed. The investigated approaches are: the pistons’ pitch angle, the cylinder ports’ pitch angle, the radial displacement of the pistons, and the radius of the pistons.

This study employs a transfer function methodology to map simulated internal pump dynamics, such as piston forces and bending moments, to audible noise. Using these transfer functions, it is possible for instance to predict how changed valve plate timing affects simulated piston forces and bending moments and in turn how this will affect audible noise. Also, different flow pulsation objectives are used in both the time and the frequency domain.

It is possible to reduce the amplitude of the flow pulsations a considerable degree when individual piston pitch angles are used; however, other objective functions increase. Neither of the approaches shows big reduction of the objective functions. However, it is difficult to rate the reduction due to the complex behaviour of the flow pulsations and transformation of the forces inside the machine. The perceived noise level may be reduced considerably despite the small decrease in the chosen objective function. In addition, the overall reduction of noise may be greater in other pump configurations than this paper shows.

Paper V

Measurement of Free Air in the Oil Close to a Hydraulic Pump

The existing noise reduction features, such as pressure relief groove and pre-compression filter volume, are more or less dependent on the working condition. It is essential to know the amount of free air when designing a quiet pump; however, it is not evident how much free air the oil contains. The free air content is different if the suction port is boost-pressured or self-priming. The amount of free air in a well-designed system can be as low as 0.5% while in others up to 10%.

This paper uses the three-transducer method to measure the amount of free air in the oil. The oil’s compressibility can be measured for different working conditions and the free air content can then be calculated. The pre-study is performed with an extensive simulation model. Various noise reduction features’ sensitivity to free air content is considered.

The paper shows that the valve plate is likely to have an impact on the
fraction of free air in the measurement pipe and therefore it is not possible to measure the outlet bulk modulus and predict the fraction of free air in the suction line.

**Paper VI**

**Unsteady Flow through a Valve Plate Restrictor in a Hydraulic Pump/motor Unit**

The motivation of this article is to investigate the reason for some particular oscillations which appear in some pump designs mainly when the pump creates heavy flow pulsations. In this article the dynamic behaviour of unsteady flow through a valve plate in an axial piston pump is investigated. The proposed extension of the steady state restrictor equation includes a dynamic internal mass term and a damping factor, i.e. inductance and resistance. The results from a 1D model is validated with a 3D CFD model. Different valve plate’s configurations and pump sizes are easily simulated with the two simulation models. The simulation results show very good comparison with experimental tests. The proposed method is verified with a hydraulic pump application but it can probably apply to original restrictors too.
References


References


On Fluid Power Pump and Motor Design


References


